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CONTENTS

page

"Not Back But Forth" 3

* * *

Sheet Metal in the Ariel "Leader" 4

When the Ariel "Leader" was first announced it was hailed as a major step forward in motor-cycle design. Sheet metal has been used to the maximum extent possible to produce a machine with great strength but light enough in weight to give a remarkable performance from a 249-cc. engine. This article describes the production of some of the main components used on the motor-cycle, including the main frame, the fuel tank, the body, the front mudguard, the leg shields, the headlamp cowling, the handlebar cover, the front forks and the side panels. Well named the "Leader" the machine retains all the advantages of the breed with the weather protection of the motor scooter.

The Local Buckling of Aluminium Plate Elements 23

K. Sutter

In this paper, which was presented to the Symposium on "The Applications of Sheet and Strip Metals in Building" organized by the Institute of Sheet Metal Engineering, the author provides all the information a designer should possess about local buckling of plate elements. Nomograms, produced as a result of much investigational work, eliminate the need for lengthy calculations and the designer has only to learn how to use them to profit from them when designing "blown-up" cross-sections on which the economics of the use of aluminium for structural members is based.

page

Sheet Metal Data Sheet ..... *facing page* 62

1—Shearing Sheet Metal

This is the first of a series of data sheets in which not only information on the fundamentals of various forming processes, etc., will be given, but also details of the equipment and machines required to effect the operations described. These data sheets are being compiled specially for "Sheet Metal Industries" by J. W. Langton, M.B.E., B.Sc. (Lond.), M.I.Mech.E.

Drum-type Pickling of Coiled Strip 66

L. W. Passmore

Some details of an interesting French Installation.

Institute of Sheet Metal Engineering 69

Forthcoming Activities.

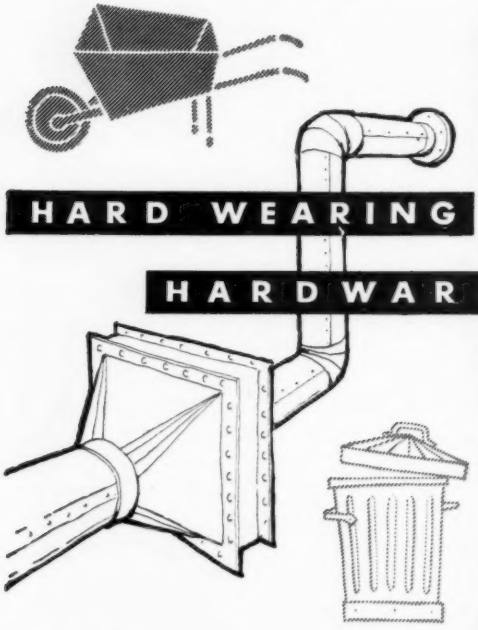
Sheet Metal News ..... 70 to 78

New Plant and Equipment 79, 80, 82, 84

Classified Advertisements 103 **Index to Advertisers 108**

A Happy and Prosperous New Year

The Editor, the Advertisement Director and the Staff extend to all readers, contributors, advertisers and other friends in the industry at home and overseas, sincere and cordial good wishes for their happiness and greater prosperity in the year that lies ahead.



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This continuous galvanising line at the Ebbw Vale works of Richard Thomas & Baldwins Limited is the largest in Great Britain. In an average week the line produces more than 1,000 tons of Speltafast strip, which is cut automatically into sheet or can be shipped in coil form. The finished material is used for roofing and ductwork, and by manufacturers of hardware, agricultural implements, and builders' equipment. The 1,000 kW Birlec furnace, which is shown in the illustration, anneals the entire output from the line. It handles various widths from 20″ up to 48″, the production rate being governed by the gauge, which may range from ·012″ to ·064″.

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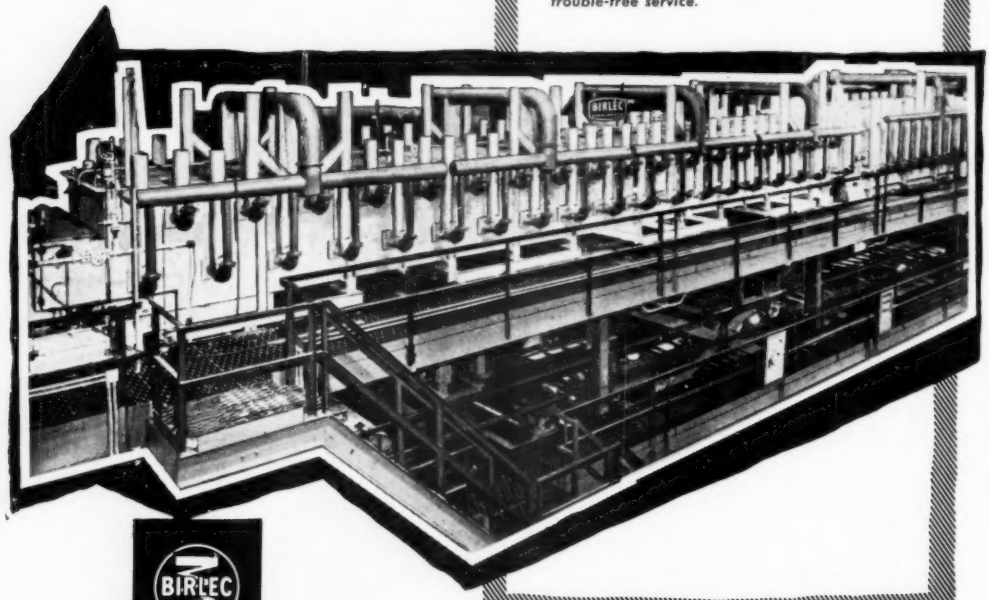

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NOT BACK BUT FORTH

THOSE of us for whom the advent of another new year has become a worrying reminder of the rapidity of the passing of time have inevitably by now developed the habit of "looking back." As we get older we tend to do this more often and in looking back we ponder on the many advances in science and engineering that took place in our younger days and often feel that today's youngsters are missing a great deal of excitement.

Further thought, however, will bring the picture into its true perspective. In days gone by major technological developments were fewer and thus each one appeared to have a greater impact. Also, the average man was not so steeped in a welter of scientific achievement as he is today, when the lay press devotes a great deal of space to this subject.

During and since the recent war, technological developments have multiplied with a speed that is quite remarkable such that any new achievement is only one of many more. However, this does not in any way reduce the value of any of them, even though their impact may not seem to be so great as it may have been thirty years ago.

In our own industry during the past year, as we prophesied, increasing attention has been paid to the use of the more refractory metals and alloys and also the use of ceramic-coated metals, for rocket and aircraft applications, exemplified by the recent announcement of the Bristol 188 in which many of these materials have been used.

Brit. research, particularly fundamental research, has always led the world, but as Sir Alexander Fleck said some time ago, "It is no use being in the forefront of research as we claim to be in this country, if we fail to apply the results," and in applying the results it is necessary to ensure that novel techniques are used in such a way as to derive the greatest possible benefit from them. Welding, for example, is a science in which a useful amount of research is continually being carried out, much of which is finding application. By many, however, welding is considered as being only one of a number of useful methods of joining metals; but in many cases the technique calls for a completely new approach to design, and this fact was highlighted during the launching of the new Design Advisory Service inaugurated by the British Welding Research Association. This service aims to inform civil and structural engineers as to how welding can be most effectively applied in terms of ease of construction and economies in material, while omitting those apparently trivial aberrations in design which can contribute to early—and sometimes catastrophic—failure.

What is true of welding is true of many other processes and practices, and although "looking back" is a pleasant occupation our thoughts should be for the future. British industry, in its practice, must continue to keep abreast of the world. Research results, and advice on how to take advantage of them, are always available and should be exploited to the full. Only in this way can continuing prosperity be assured.

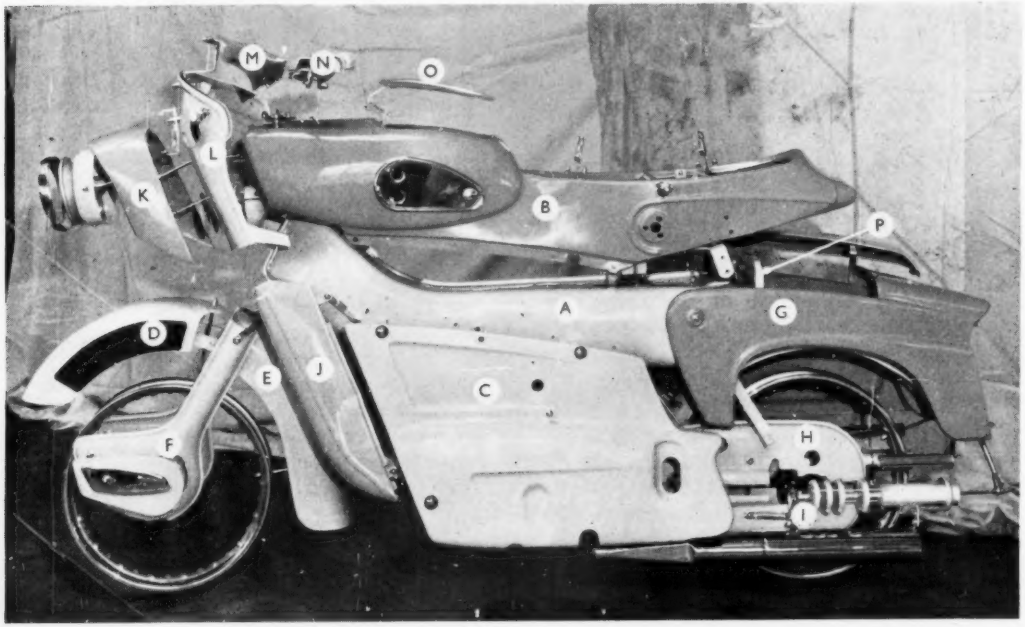
Sheet Metal in ARIEL

A New Concept in Motor Cycle design . . .

Introduction

WHAT is undoubtedly a most significant step forward in design as far as British motor cycles are concerned was reached in 1958 with the introduction of the Ariel "Leader." Powered by a 249-cc. air-cooled two-stroke engine, the machine is capable of 70 m.p.h., a performance due in great measure to the fact that the main components of the cycle, *i.e.*, the main frame and outer body shell, are of fabricated sheet steel construction (Fig. 1). The machine is almost completely enclosed, but side panels are readily removable to give first-class accessibility to the power unit and transmission. The design of the frame and body is such that there is ample parcel accommodation; the battery and a tool locker are also enclosed within the body; the chain is enclosed in a case composed of two easily removable pressings and the rear end of the body and the rear mudguard hinge upwards to give access to the rear wheel.

Fig. 1.—*Some of the component parts of the Ariel "Leader":* (A) *Main frame;* (B) *body;* (C) *side panel;* (D) *and* (E) *front mudguard;* (F) *front-fork leg;* (G) *tail cover;* (H) *chain case;* (I) *exhaust/silencer assembly;* (J) *leg shield;* (K) *headlamp cowl;* (L) *front shield;* (M) *instrument panel;* (N) *handlebar cover;* (O) *glove-box lid;* (P) *bulkhead (fitted inside frame at rear of petrol tank)*



the "LEADER"

By L. J. Bacon

The machine also incorporates a cowled headlamp (Fig. 2), a front shield continued downwards on each side into leg shields, an instrument panel, pressed-steel front forks brazed into a malleable iron steering crown, and a pressed-steel cowling for the handlebars.

The main frame comprises, in general, two 20-gauge steel pressings welded together down a centre-line flange to form a rigid box-section structure (Figs. 3 and 5). The frame is upswept at the front to allow the steering column to fit into the steering head at an angle of 65 deg. The frame cross-section is rectangular with radiused corners, and the sides are bowed to resist drumming. The steering-head end of the frame is slightly tapered and at the rear it widens and is forked to form the anchorages of the upper ends of the rear suspension spring units.

K. J. Whistance, Managing Director, Ariel Motors Ltd.

Fitted inside the frame is a 2½-gal. fuel tank (Fig. 4) composed of two identical 20-gauge pressings welded together. Behind the tank a perforated and ribbed bulkhead is bolted in.

Three box section brackets welded underneath the frame (*see* Fig. 3), also fabricated from 20-gauge sheet, are used to support the engine. The forward two are so positioned that most of the load is taken in shear, and these two brackets are braced by a welded-in channel section bridge piece. The rear bracket, a single unit, incorporates two sleeves, the upper one for the engine attachment bolt and the other for the rear fork spindle. This bracket is also used for an air-silencer chamber, and a stub projecting from the bracket is used for a replaceable air filter.

The rear fork is of welded D-section tube; and below this is a roll-on stand, carried in two engine-unit lugs.

The deep front mudguard is made in two halves with guttered edges, the two halves overlapping at the fork stanchions to which the halves are sealed with rubber beading. To the inner pressing of each fork stanchion is welded a curved channel-section bracket with the flanges facing outward. Welded to the inside of the valances of each mudguard are similarly curved top-hat sections which slide over the brackets. Each mudguard is fastened by two bolts which screw from the inside into captive nuts, and one upward into the fork yoke.

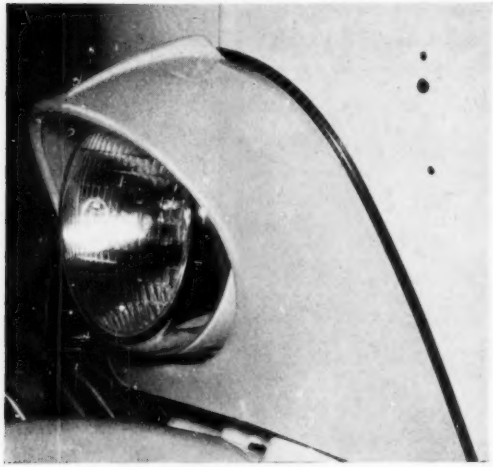


Fig. 2 (*above*).—*Close-up of headlamp cowl*

The body shell that encloses the main frame comprises eight main pressings in 20-gauge steel, *viz.*, the upper mid section, tail portion, two side panels, two lower legshields, frontal fairing incorporating the legshield upper ends, and the headlamp cowling. The upper mid section is itself a welded fabrication of five main pressed components. This mid section is bolted to the main frame, and its forward section is shaped rather like a conventional petrol tank that fits over the steering head. On top of this section behind the steering head is a lockable pressed steel hinged cover giving access to the parcel locker.

The hinged dual seat gives access to the rubber-mounted tool kit and battery, etc., and the opening on which the seat rests is braced by a transverse tube that houses a pull-out lifting handle. The battery is suspended by a U-shape steel strap.

The body tail section (Fig. 6) is attached to the mid-section by a pivot bolt at each side; the side

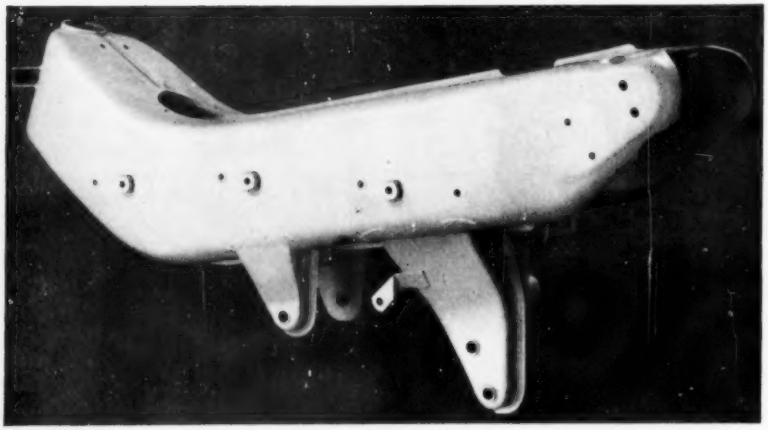


Fig. 3 (*right*).—*Main frame complete with engine-mounting brackets*

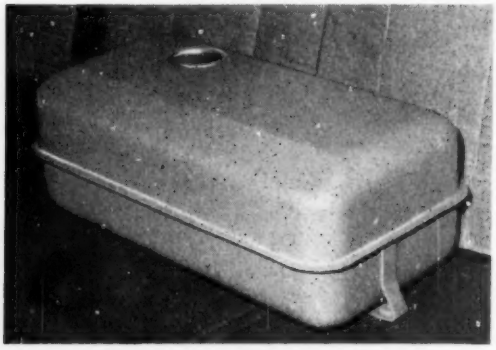
panels that enclose the power unit are attached to the body by five captive screws with coin slots. The frontal fairing is held to the mid-section by seven bolts, and the upper ends of the legshields fit into channels formed at the base of the fairing. The shields are attached to the main frame at the top and to brackets bolted to the crankcase; these brackets also form the lower front end pick-up points for the side panels.

Other sheet-metal components include the instrument panel and a cowling that fits over the handlebars (*see* Fig. 7). Pressed steel panniers can be supplied if required.

The total weight of the machine is about 300-lb.

The production and assembly of the pressings for the " Leader " is truly a co-operative effort on the part of several firms, including A. J. Homer and Sons Ltd., Fletcher Bros. (Pressings) Ltd., Midland Motor Cycle Mudguard Co. Ltd., and William

Fig. 4 (*below*).—*Petrol tank*



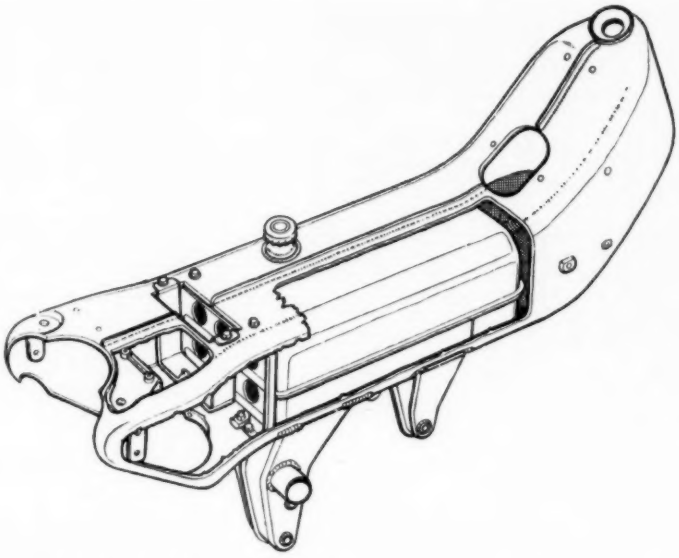


Fig. 5 (left).—General drawing showing construction of main frame cut away to show interior and position of petrol tank

Fig. 6 (below).—Tail-cover assembly

Fig. 7 (below, left).—View of instrument panel and handlebar cover

Pearce (Pressings) Ltd., and for the production of some of the larger pressings a 300-ton Wilkins and Mitchell press was specially installed by A. J. Homer.

As the design of the machine was brought to fruition, it was found possible further to extend the use of sheet metal, for example, for the brake shoes (Fig. 8) and for the clutch unit (Fig. 9), for parts of the front suspension unit, etc. In addition, the silencer assembly is typical of the more conventional use of sheet metal on a motor-cycle (Fig. 1).

Production of Pressings

A. J. Homer and Sons Ltd., Shirley, who are internationally known as manufacturers of motor-cycle and other types of petrol tanks, have a major role in the production of the "Leader." They manufacture the petrol tank, the front forks up to the headstock, the main frame complete, and the bulkheads.

Petrol Tank (*Figs.* 4 *and* 10)

Produced from 20-gauge extra-deep-drawing quality steel manufactured by John Summers and Sons Ltd., the petrol tank is pressed in two identical halves on a Taylor and Challen No. 6 double-

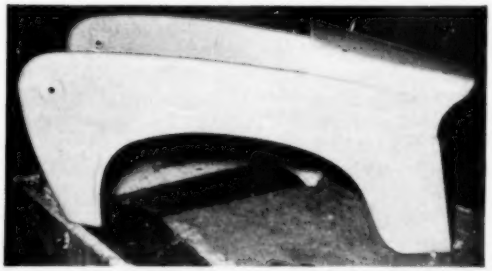


Fig. 8 (right).—Brake shoes

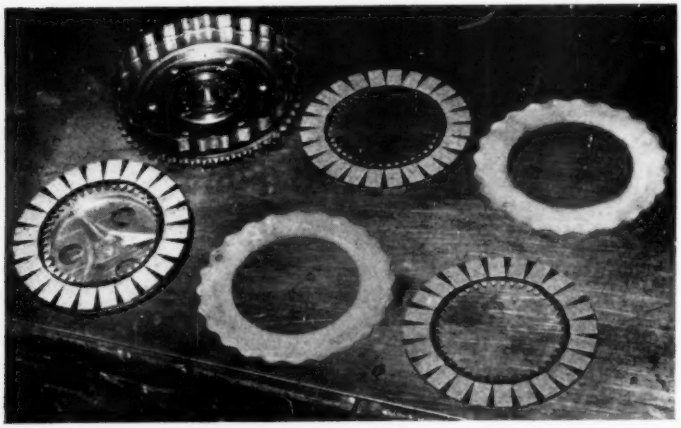


Fig. 9 *(above).—Component parts of clutch assembly*

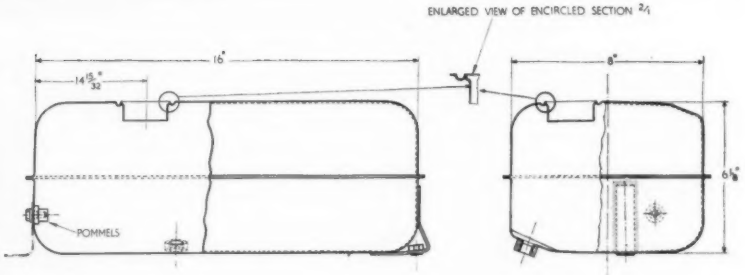


Fig. 10 *(right).—Dimensioned drawing of petrol tank (extra-deep-drawing steel* 0.031-*in. thick)*

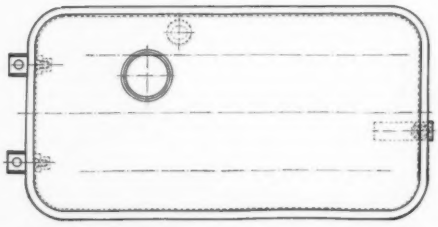


Fig. 11 *(below).—Completed fork legs before brazing to steering crown*

action press, the sequence being—blank, raise and crop, effected as separate operations. After pressing, the two halves are mounted in a jig and spot-welded together round the flange. Spot-welding is only a "tacking" operation, the final weld round the flange being made by a 100-kVA. British Federal seam welder. This machine has an inclined wheel so that during the seam-welding operation the wheel can clear the front tank mounting bracket attached by spot welding to the bottom tank pressing before the two halves are tack spot welded. During the pressing operation pierced holes are produced for the filler neck and the petrol tap boss. The filler neck is itself a pressing gas welded into the top of the tank; the petrol tap boss is projection welded (Sciaky 200-kVA. machine) into the bottom of the tank. Also during pressing, holes are pierced to take two threaded bosses, welded to the inside rear end of the tank, to which a right-angled bracket containing four holes is bolted. These brackets form two out of the three mounting points for the tank. A similar boss is incorporated in a simply formed bracket, spot welded to the forward end of the bottom tank pressing; this is the third attachment point for the tank. The completed tanks are

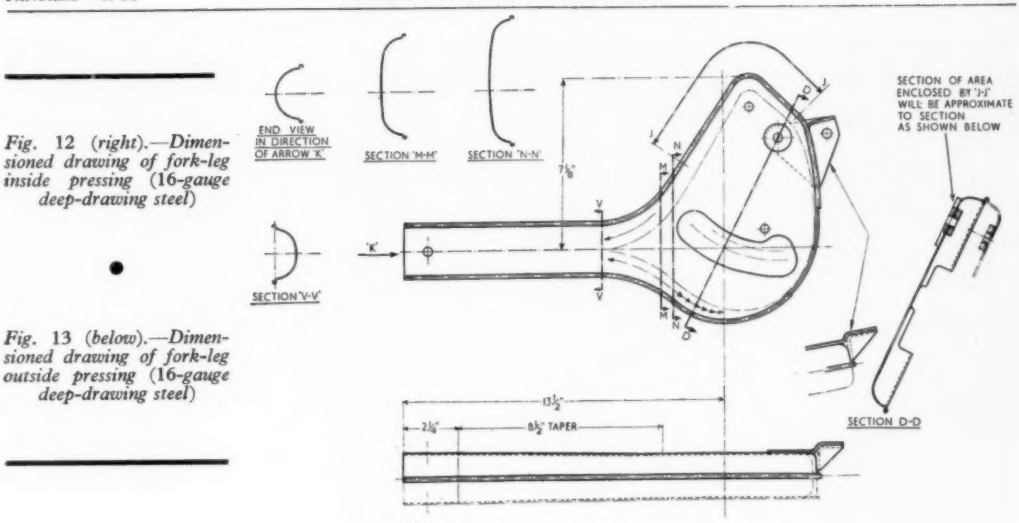
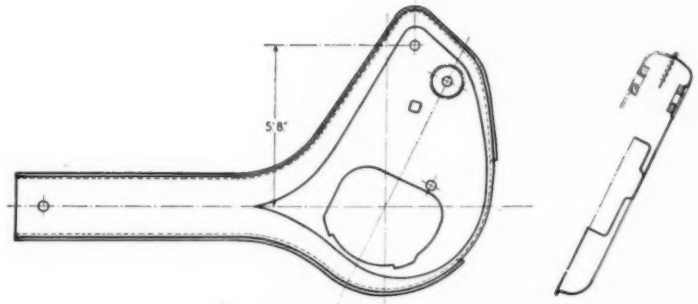


Fig. 12 (*right*).—*Dimensioned drawing of fork-leg inside pressing* (16-gauge *deep-drawing steel*)

Fig. 13 (*below*).—*Dimensioned drawing of fork-leg outside pressing* (16-gauge *deep-drawing steel*)



Bonderized and coated with grey primer. The threaded bosses are all projection welded into the tank by means of a 100-kVA Sciaky machine.

Fork Legs (*Figs.* 11, 12 *and* 13)

The two front fork legs are produced from two identical pairs of pressings, each pair being obtained from a single 16-gauge blank (produced in a 150-ton Rhodes press from Summers' E.D.D. steel) which is cropped and parted as a final operation. The large apertures in the leg components which give access to the suspension units and which are covered on the completed machine by a removable plate, are pierced in a separate operation on a Sweeney and Blocksidge 30-ton power press; the slot in the bottom of the leg is pierced on a hand press. The boss on the inside of each leg pressing, in which the front suspension pivots, is projection welded in place by a Sciaky welder. This boss is machined to size in a subsequent operation at the Ariel works.

One half of each leg has a larger flange than the other (produced during blanking on the Wilkins and Mitchell 300-ton press). The wide flange is turned through 90-deg. and the two halves of each leg are then jigged and spot tack welded together. The flange is then closed over in two operations (Fig. 14) on a Taylor and Challen No. 3½ press; the first operation turns the flange through a further 45 deg., the second giving the final closing.

The large aperture in each leg is for fitting and adjustment of the front suspension unit.

Rear Engine Mounting Bracket

This component is first blanked in the form of the two joined halves of the component, and the required holes are produced in a combination pierce and plunge operation, using a Taylor and Challen No. 4½ double-action press, while the two halves are still in one piece. The two halves are then separated. The air cleaner which is incorporated in the bracket is formed up and gas welded, and then welded into the bracket. The complete bracket is assembled on a jig consisting of two vertical rods which square up the two sets of pressings; the tubes (including the air cleaner) are slipped on the rods between the two pressings and the whole assembly spot welded together and then roller seam welded (*see* Figs. 15 to 17).

The flanged top of the bracket, used to attach the bracket to the frame, is now produced as a separate component which is then forced, in a jig, on to the main bracket assembly and then spot welded. The top portion is drawn, using a Taylor and Challen No. 4½ press, from a shaped blank to save cropping.

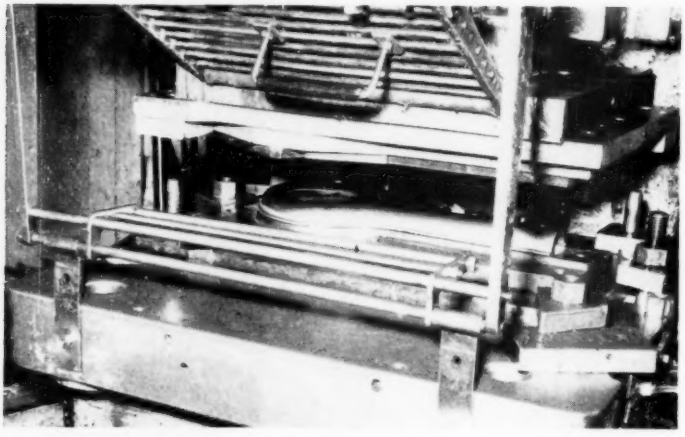


Fig. 14 (*left*).—*Fork leg pressings in press for closing over of flange after spotwelding*

Fig. 15 (*below*).—*Drawing of rear engine-mounting bracket which incorporates the air cleaner* (20-*gauge mild steel*)

Front Engine Mounting Brackets

Again, the two halves of each bracket are blanked in one piece and pressing carried out, after which the whole is cropped and then parted. Holes are combination pierced while the two halves are still joined. A Taylor and Challen No. 4½ press is used. The centre section of this bracket is blanked (Taylor and Challen No. 3½ press) and two boxing operations follow. Holes are pierced singly and plunged using a hand press. The bracket is spot and seam welded up

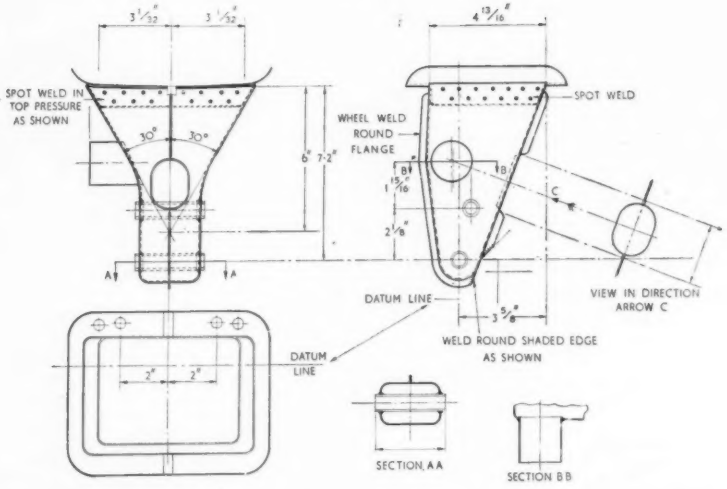
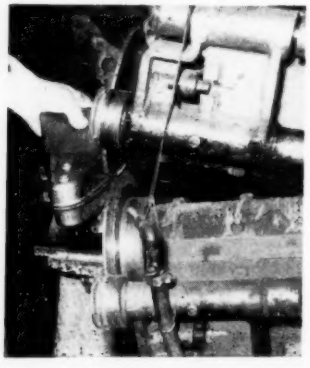


Fig. 16 *and* 16*a.*—*Method of assembling rear engine-mounting bracket*

Fig. 17.—*Roller seam welding two halves of rear engine-mounting bracket*



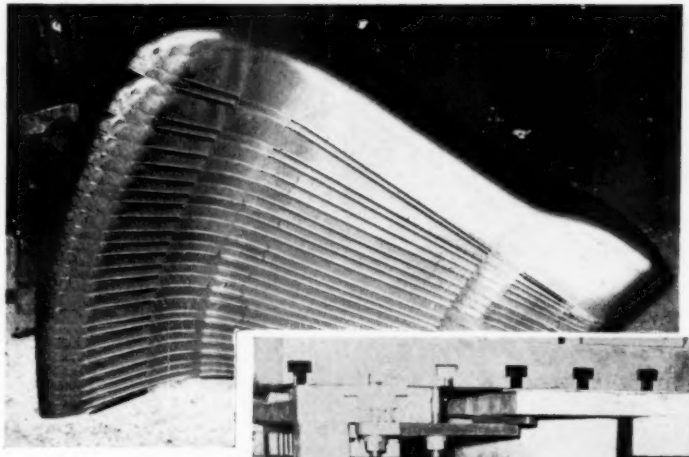


Fig. 18 (*left*).—*Half pressings of main frame*

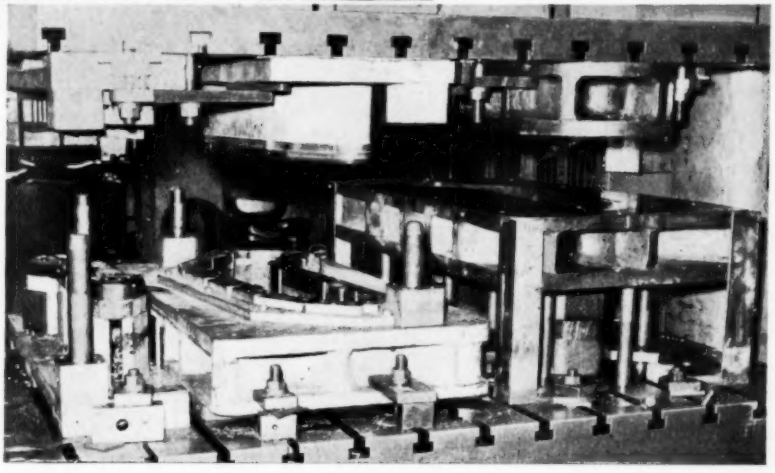
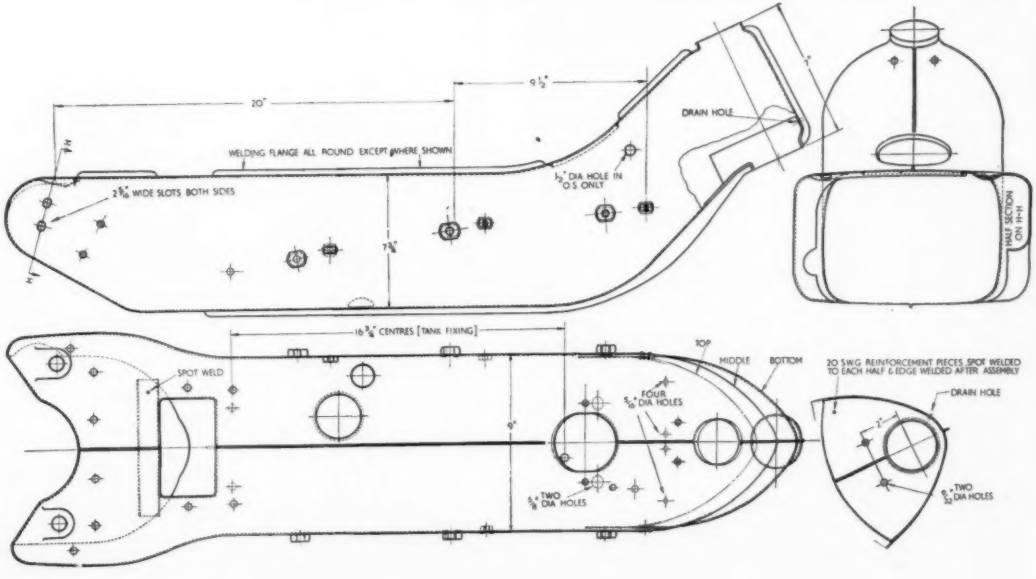


Fig. 20 (*below*).—*Press tools for half pressing of main frame*

Fig. 19 (*below*).—*Dimensioned drawing of main frame material* 20-*s.w.g. steel* (*deep drawing*). *The Quasi-Arc Co. Ltd.'s "Vortic"* 16-*gauge electrodes are used in the fabrication of the frame*



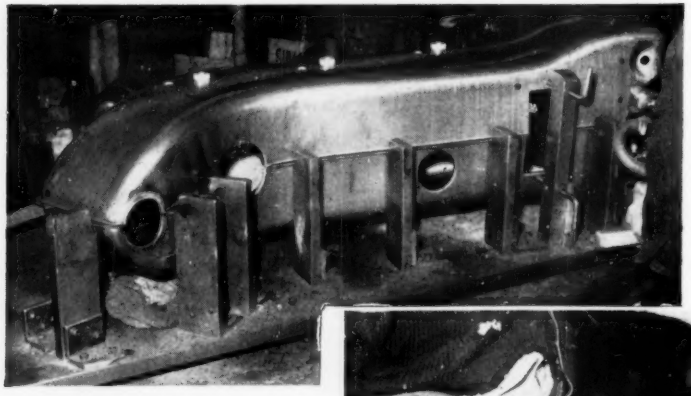
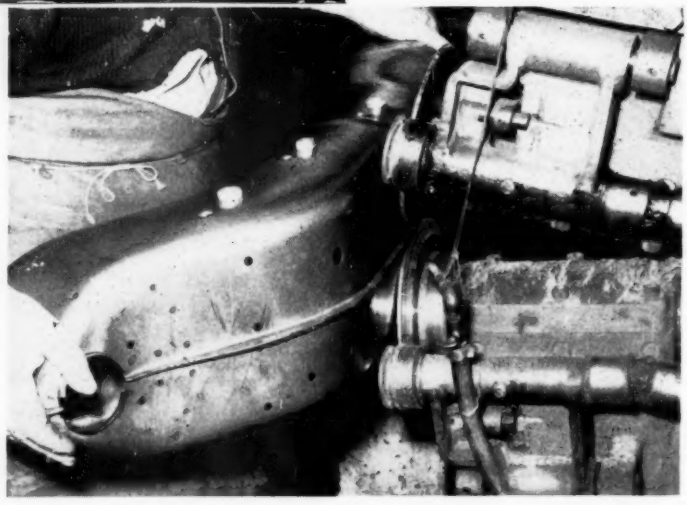


Fig. 21 (left).—Main frame pressings jigged for tack spotwelding together

Fig. 22 (below).—Roller seam welding of main frame pressings



and the tubes forming the two engine mounting holes are gas welded in. These two brackets when attached to the frame are separated by a formed-up channel section piece.

Frame

The two halves of the main frame are massive 20-gauge pressings (Figs. 18 and 19) and necessitated the installation of the 300-ton Wilkins and Mitchell press. Its bed accommodates three sets of tools for raise, crop and com-

Fig. 23.—Jig for assembling engine-mounting brackets to main frame

Fig. 24.—Tack spotwelding engine-mounting brackets to main frame

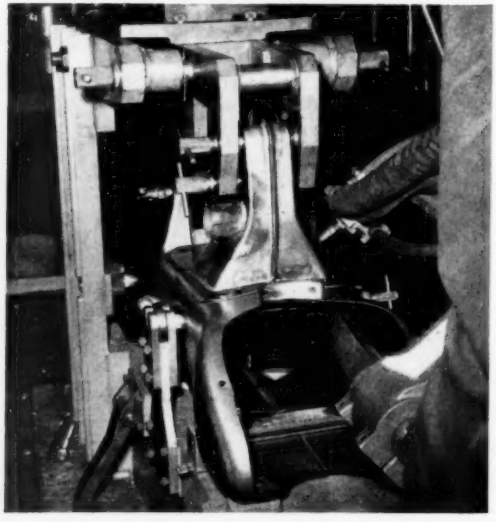
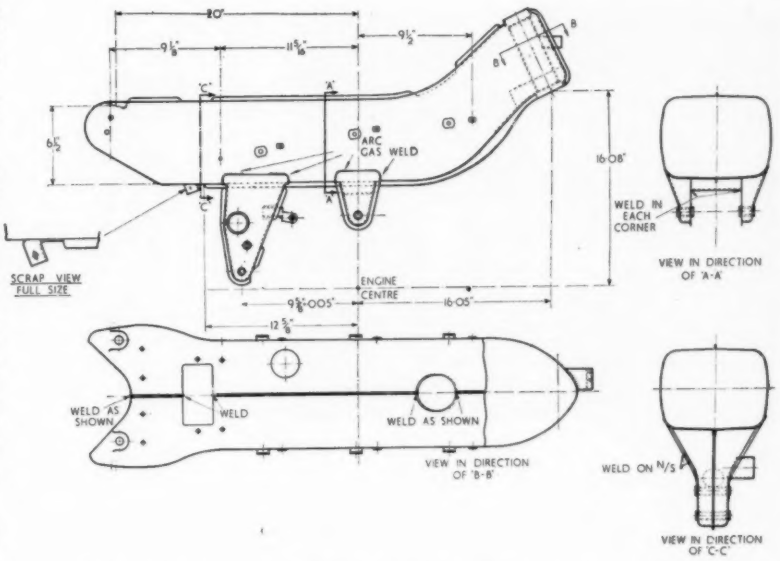


Fig. 25 (right).—Drawing of frame welding assembly



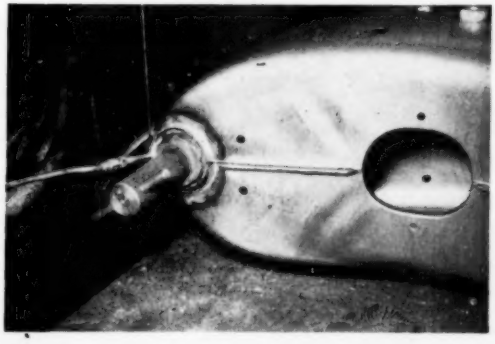
bination pierce operations on previously produced shaped blanks.

Assembly of the frame is by jigging the two halves, spot tack welding round the flange (Fig. 21), and finally joining by roller spotwelding (Fig. 22). A reinforcing piece is welded into the front end to stiffen up the steering head. The rear stiffeners are then added and also the mudguard bracket. The assembly is then jigged using the body attachment holes, and the engine mounting brackets are placed in the jig using the mounting holes as a location (Fig. 23). The front and rear engine mounting brackets are then spotted on to the frame (Fig. 24). The frame welding assembly diagram is shown in Fig. 25.

The assembly is then jigged again, and location this time is by the engine mountings. At this stage the steering head tube is welded into the frame (Fig. 26) and the horn mounting bracket gas welded to the front of the frame.

The steering head assembly consists of a gas-welded 16-gauge sheet-metal formed tube, on to the top and bottom of which a recessed pressing is gas welded. These pressings take the machined cups which form part of the steering head ball races. The head assembly is inserted from the bottom of the frame and when in position is gas welded to the top and bottom of the frame without using filler rod. Enough metal is allowed on the flange round the holes in the frame to allow for this. Thus the top side of the frame involves the welding of two thicknesses of metal, the frame itself and the pressing on the head assembly, while at the bottom three thicknesses are involved

Fig. 26 (below.)—Gas welding steering head tube into main frame



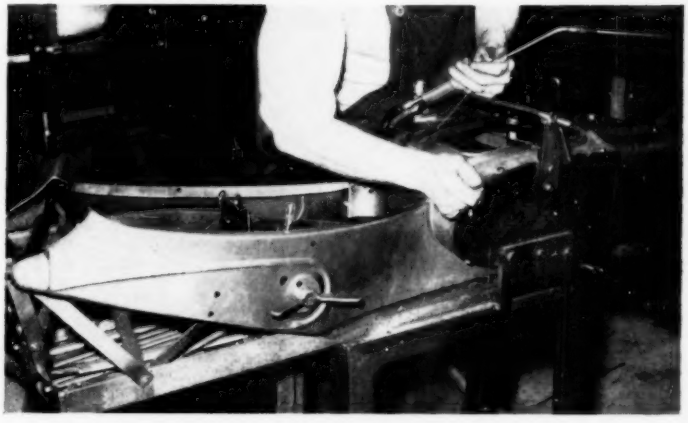


Fig. 27 (right).—Gas-welding operations on jigged body pressings

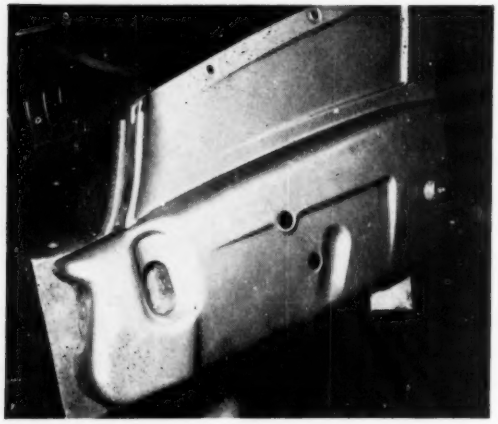


Fig. 28 (above).—Side panel set up for planishing

producing high-quality pressings for the motor and allied trades for many years.

The components produced for Ariel Motors Ltd., comprise :—

(i) Body side panels ;
(ii) Headlamp cowls ;
(iii) Front shields ;
(iv) Legshields.

Body Side Panels (Right and Left Hand)

These panels (Fig. 28)—formed from 22-gauge material—provided a problem in the early stages. No double-action presses were available, and a Wilkins and Mitchell press No. 844, equipped with air cushions, was eventually used for the job. Although the dies were fitted with draw-beads, a great deal of experimenting with blank sizes, material specifications, cushion pressers, etc.

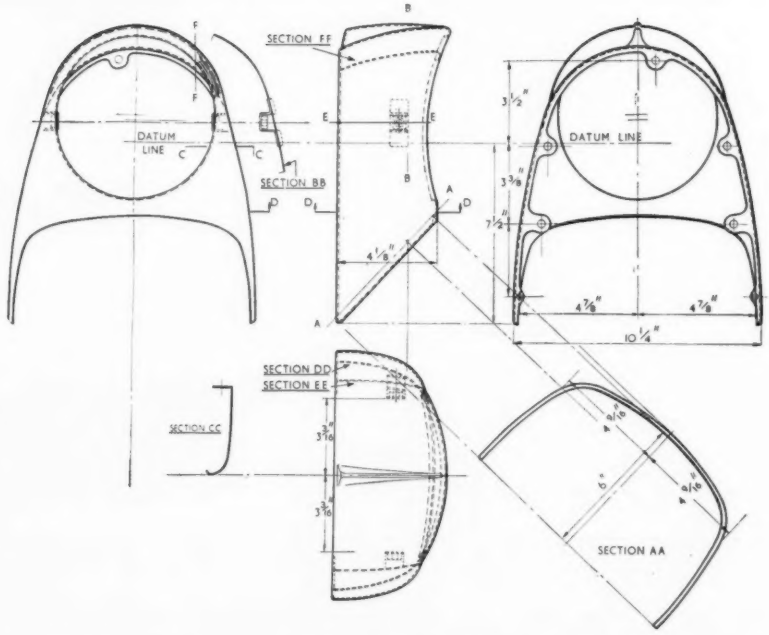


Fig. 29 (right).—Drawing of headlamp cowl (22-gauge deep-drawing steel)

due to the stiffening plate welded into the frame, as previously described.

The body of the machine which is assembled by A. J. Homer consists of eight main components assembled in jigs (Fig. 27) at first by gas and spot tack welding. Final welding is by spot welding at the Ariel works. These body pressings are produced by Fletcher Bros. (Pressings) Ltd., Holyhead Road, Handsworth, Birmingham 21, who also produce other pressings for the " Leader " motor-cycle. This company is a member of the Concentric Group of Companies, and has been had to be carried out before the panels could be successfully formed to full depth in one operation.

All the usual troubles, *i.e.*, cracking, puckering, crowding of metal into corners, etc., were successively met with and overcome, and the best results are now being obtained by using stabilized material.

The gauge thickness is closely controlled, and R. D. Nicol's " Droyt " drawing compound is used.

It is also interesting to note that a small piece of coarse emery paper placed on the blank was the answer to the problem of metal being drawn from several directions and converging at a point which

made crowding of the material almost inevitable, and—although considerable experimenting was carried out with draw-beads—the use of emery paper provided the complete answer.

Subsequent operations included the piercing of the majority of the numerous holes in one operation, and clipping operations to get rid of the surplus material and finalise the shape of the pressings.

All the tools were produced in the firm's own tool room, and because of the fairly modest output required, easily maintained tools were provided.

Headlamp Cowl (*Figs. 2 and 29*)

The headlamp cowl is also an interesting pressing, and initially the chief difficulty was to provide a good and presentable finish to the top of the cowl, *i.e.*, the portion which the rider would see when seated in the saddle.

Fig. 30 (*above*).—*Interior view of leg shield*

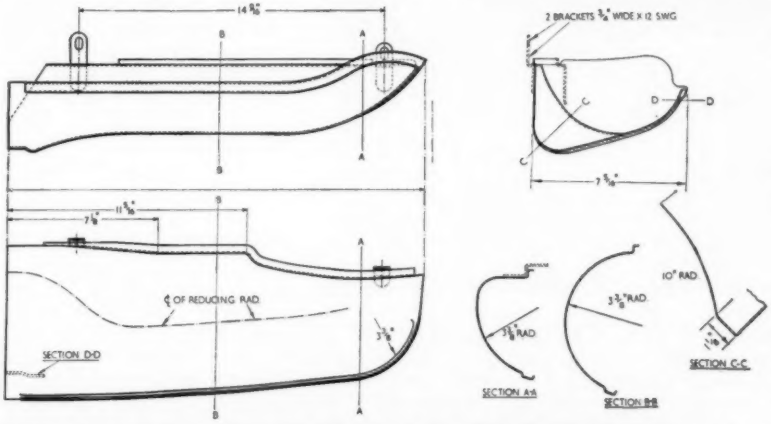


Fig. 31 (*left*).—*Dimensioned drawing of leg shield (22-gauge deep-drawing steel)*

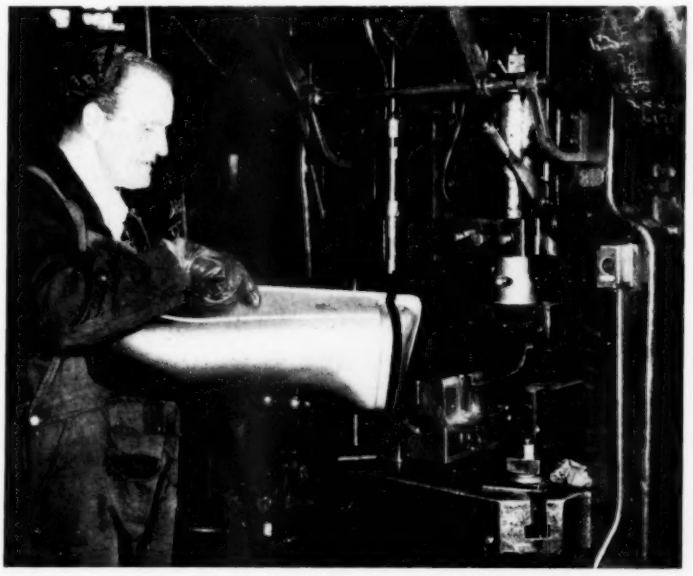


Fig. 32 (*below*).—*Clipping operation on sides of leg shield (Sweeney and Blocksidge No. 38 press)*

The rib was not included, and it can readily be imagined that the combination of the two large radii—running at right angles to each other—resulted in puckering and crowding of the metal.

The method originally used was to notch out a V-shaped piece of metal and close the edges together to provide the contour called for by the drawing, but as welding and pressing operations were involved, the method proved very costly.

In consultation with Ariel Motors Ltd., it was decided that pushing up the surplus metal to form a rib would not

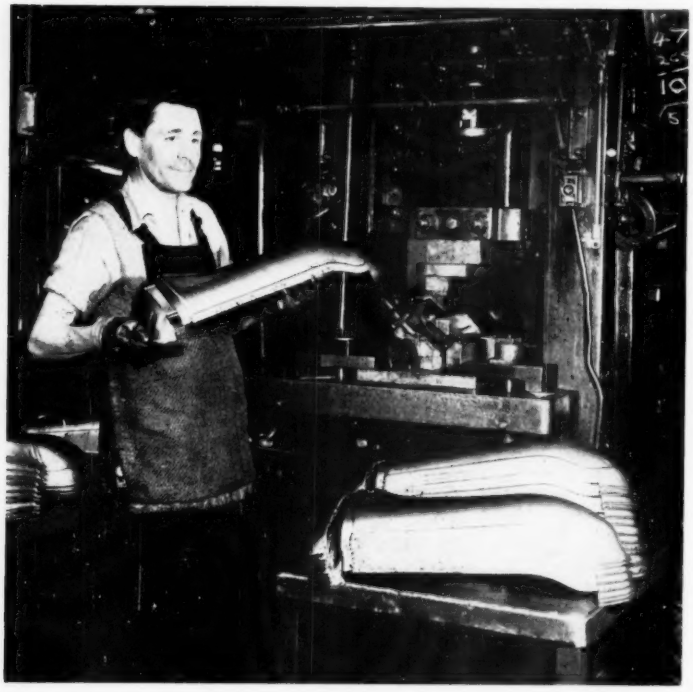


Fig. 33 (left).—Clipping operation on bottom of leg shield

Fig. 34 (below).—Dimensioned drawing of handlebar cover (20-gauge deep-drawing steel)

only improve production, but also result in a more streamlined and pleasing appearance.

Front Shields and Legshields (*Figs.* 30 *to* 33)

The front and leg shields presented no undue difficulties to produce, although it was found necessary to use stabilized material, and pay particular attention to gauge thickness to ensure that the pressings were free from " kinks " and surface blemishes.

Another firm invited into the team chosen to produce components for the "Leader" was William Pearce (Pressings) Ltd., 44, Ormond Street, Birmingham, 19.

A firm of wide repute in the car and motor-cycle manufacturing industries, about 120 staff are employed, and in the 25 years the company has been in existence, a considerable amount of experience and " know-how " of the complexities of metal forming have been built up.

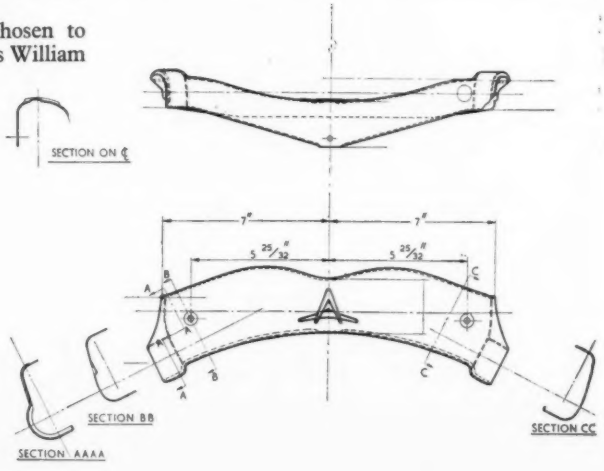
Pearce's undertook to provide 47 components for the machine, and to produce these, approximately 450 sets of tools have been designed and manufactured by the company itself. To meet the time schedule initially drawn up for production samples, the company's tool room developed the tools within nine months, and not one of the samples subsequently submitted was rejected by Ariel Motors Ltd.

The main components being made by Pearce include the instrument panel, handlebar cover, and clutch housing (*see* Figs. 7, 9 and 34). When the original cast brakeshoes were later replaced by others fabricated from heavy-gauge sheet steel, these also were made by the company (Fig. 8).

The company are currently building a new factory in the Brownhills area of Birmingham for presswork and final assembly, and the main press shop is to occupy 7,000 sq. ft. Office and store accommodation, laid out on modern lines, will complete what will be an attractive and efficient plant.

Instrument Panel

The material used for this component is 22-gauge deep-drawing-quality mild-steel sheet with full



finish. The component was too large to be formed on the firm's existing double-action presses, and a Taylor and Challen Model 1664 100-ton open-fronted press was used. Despite this apparent fitting of a " quart into a pint pot," a satisfactory component was produced.

Brief details of the Taylor and Challen 1664 press are as follows :—An open-fronted geared press, the bed and slide are of large area, and both have machined T-slots. The slide is spring balanced, and ample tool space, combined with rigidity of frame, make the press useful for a variety of comparatively heavy press operations. A patent automatic key clutch and one-stroke trip gear are fitted as standard. The machine will cut out a blank in mild steel 19 in. dia. by $\frac{3}{32}$ in. Extra equipment obtainable includes top and/or bottom extractors, spring or pneumatically-operated die cushion (for use with combination tools), and an adjustable stroke giving a range of $\frac{1}{2}$-in. to 4-in. or $1\frac{1}{2}$-in. to 5-in.

The stroke maximum is 7-in. (usual 4-in.) with a usual rate of strokes per min. of 35. Bed area, right to left, is 44-in., and front to back 26-in.

Made from a sheared blank, the panel was first raised and embossed on its top section, then clipped and redrawn. Side cropping and piercing tools were employed for follow-up operations, so that hand work has been virtually eliminated in production.

After tooling had started, the panel section was deepened by a considerable amount, and it was at this stage that it was discovered that the component was too big for the double-action press initially scheduled for the job. The tools were adapted to the Taylor and Challen press with successful results.

Handlebar Cover (*Figs. 7 and* 34)

Produced from the same gauge and quality sheet as the instrument panel, a developed blank was used for the handlebar cover to eliminate a number of cropping tools that would otherwise have been required.

Most of the larger raising tools used in manufacture were produced from nickel-chrome castings, cast to close size control, and the company says that this enabled considerable economies in tool costs over conventional methods.

The difficulties experienced in making the cover were caused by crowding of the metal into the centre section, and skidding during the raising operation, particularly since no tooling holes were available for use as locating points.

Crowding was overcome by incorporating a raised letter " A " as an embellishment on the top face of the cover. A developed blank was reverted to when it was found that a balancing effect created by increasing the blank on the short side led to the crowding reaching impossible dimensions. With a developed blank the embossed " A " was sufficient to overcome this difficulty. Semi-combination type tools were used for the raising of the panel.

Skidding commonly experienced with this type of raising varies considerably, of course, when a discrepancy in the nominal gauge of the metal is encountered, a variation of 0.003-in. being sufficient to make the job impossible.

A pre-bending operation introduced before the raising procedure largely overcame this problem.

The sequence of operations finally adopted for making the cover was, therefore: forming from a developed blank, pre-bent, raising, forming and piercing the back, and turning in the ends to fit in the twistgrip groove of the handlebar.

Clutch Housing

The clutch housing (Fig. 9) is a two-dimensional diameter shell of 16-gauge sheet steel. The open

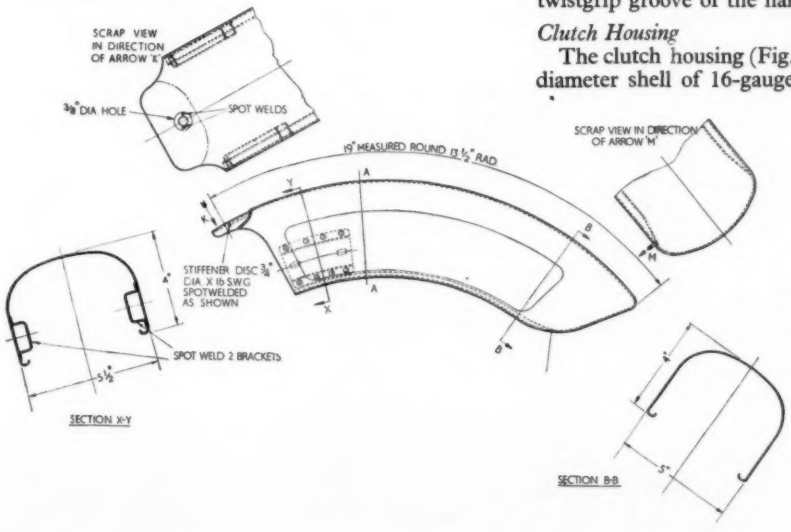

SCRAP VIEW IN DIRECTION OF ARROW 'X'
3/8" DIA HOLE
SPOT WELDS
19" MEASURED ROUND 13½" RAD
SCRAP VIEW IN DIRECTION OF ARROW 'M'
STIFFENER DISC 3/8" DIA X 16 SWG SPOTWELDED AS SHOWN
SPOT WELD 2 BRACKETS
5½"
SECTION X-Y
SECTION B-B

Fig. 35.—*Dimensioned drawing of front mudguard leading section.* (*E.N.2A, 22-gauge*)

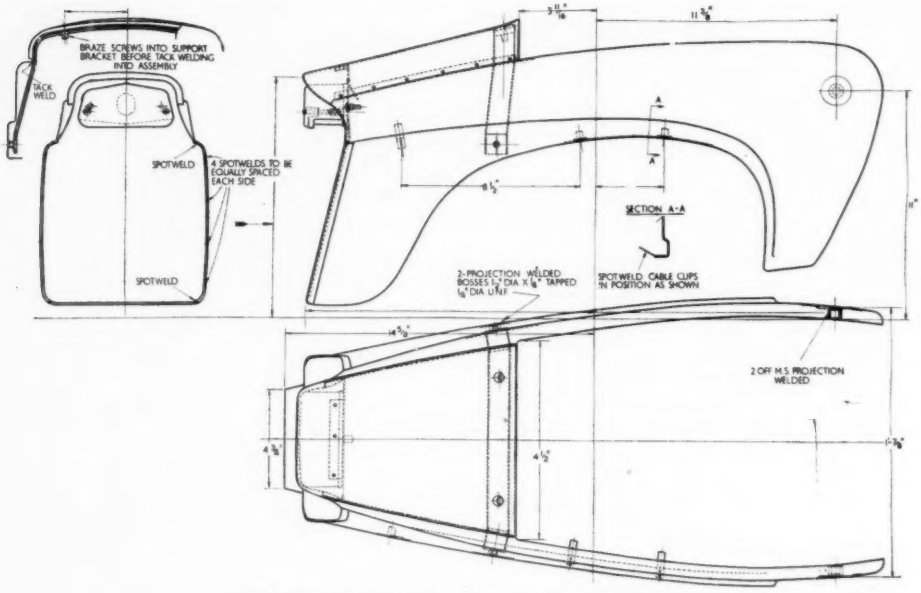

BRAZE SCREWS INTO SUPPORT BRACKET BEFORE TACK WELDING INTO ASSEMBLY
TACK WELD
SPOTWELD
4 SPOTWELDS TO BE EQUALLY SPACED EACH SIDE
SPOTWELD
SECTION A-A
2-PROJECTION WELDED BOSSES ¼" DIA X ⅛" TAPPED ⅛" DIA UNF
SPOTWELD CABLE CLIPS IN POSITION AS SHOWN
2 OFF M.S. PROJECTION WELDED

Fig. 36.—Dimensioned drawing of tail cover assembly

end of the large diameter is scalloped around the periphery to match corresponding tongues in the clutch plates.

Using a formed blank, the raising operation is made with semi-combination tools, and the housing is subsequently blocked to impart the critical squareness and flatness required for the plate bearing area.

★ ★ ★

Another firm making components for the "Leader" is Midland Motor Cycle Mudguards Ltd., Sandy Lane, Birmingham 10, founded only six years ago as an associated company of Bacol Manufacturing Co. Ltd., a firm specializing in the manufacture of press tools and presswork.

In this comparatively short time the company has expanded rapidly, and additional premises have recently been acquired to contend with increasing demands.

The company is adept at devising tools for the more unorthodox type of presswork. Prototype examples of intricate forms that have recently been made by them include a front mudguard of modern design that incorporates a deep sectioned body fairing, and a cowling that provides not only a shroud for the top section of a telescopic fork, but also a housing for a headlamp unit.

Three major components for the "Leader" being made at the company's works are the front mudguard, the rear chaincase, and the tail cover assembly.

Front Mudguard (*Fig.* 35)

This comprises two pieces—a front and a rear—formed from high grade 22-gauge steel.

Many operations are involved in the manufacture of these guards, from an initial rolling to form a straight section carried out in one operation on a rolling machine; this utilizes six sets of precision rolls driven by a 25-h.p. motor through a reduction gear. After train rolling the section is passed through a curving machine, comprising top and bottom precision rolls and a slipper or former that can be mechanically adjusted to give the required radius. Any radius can be obtained by adjustment of the slipper. The drive to the machine is from a 7½-h.p. electric motor through a chain.

All-special purpose equipment and precision rolls for these operations were made by Midland Motor Cycle Mudguards.

Tail Cover Assembly (*Figs.* 6 *and* 36)

The tail cover assembly is fabricated from four pressings, all of 22-gauge d.d.q. mild steel, comprising a left- and a right-hand side panel, a top cover, and a number plate. The four components are spot welded together, and a raised strap in 10-gauge steel is brazed between the side panels and top cover to provide a rear anchorage for the component on the motor cycle.

Used in the production of these parts are a Hordern, Mason and Edwards 200-ton double-sided double-crank press, two open-fronted presses,

a Taylor and Challen 100-ton and a Bliss 75-ton, and No. 5 fly presses.

In the pressing of the panels, buttons inserted in the punch ensure that the flow of metal around the compound curvature at the rear of the panel is smooth and free from wrinkling.

After the initial forming of the top cover, the workpiece is mounted on a mandrel in a No. 5 fly press. An interesting tool developed by the company is that used to impart the final forming of the turned over flange. It comprises two mechanically operated side-action dies that clamp the workpiece to the mandrel before the vertical force is applied, ensuring that the correct form of the piece is maintained.

Joining of the parts begins with the brazing operation, when stud bolts are welded to the 10-gauge strap, the strap is brazed to the top cover, and threaded bosses are brazed into position in each of the holes in the side panels.

After this, a jig is used for the spot welding operations, when the number plate is joined to the side panels, and the top cover is joined to both side panels and the number plate. The jig aligns with the pivot holes in the side plates and the weld-studs on the brazed-in strap.

Welding machines used for these operations are a Holden and Hunt 15-kVA. and a British Federal 8-kVA. resistance welding machines. Brazing is with " Sifbronze " No. 1 spelter rod, supplied by the Suffolk Iron Foundry.

Rear Chaincase (*Figs.* 1 *and* 45)

The component is designed as an enclosure for the rear driving chain, and is not intended to be oiltight. Although the finished case consists of a top and bottom half, it is made in two pieces as an inside and an outside. Each piece is subsequently divided along a longitudinal centre line, the two bottom halves being then joined, and the two top halves joined, by a standing seam, spot welded around the periphery.

The main operations in its fabrication comprise blanking and forming on 200-ton double-sided double-crank presses. Included in the raising are raised portions to clear the swinging arm and rear plunger, and a joggled strip that is subsequently cut through to form a stepped joint.

Following operations comprising : (1) piercing eight apertures in the front section and six in the rear ; (2) turning the flange on the front section through 90 deg. ; (3) dividing longitudinally ; (4) completing the standing seam to make the upper and lower sections ; this is done by first bending the flange through a further 45 deg. on a seaming machine, and then making the final turnover by using a punch and die conforming to the profile of the case halves in 75-ton open-fronted presses ; and (5) five spot weld spaces around the periphery of each section to complete the joining operation.

The finished chaincase can be assembled to the motor-cycle without the need for any further work in fitting.

Throughout all drawing operations at the company's works, the lubricant used is Edgar Vaughan and Co.'s " Hough to Draw."

Final Assembly

At the Ariel works, Selly Oak, Birmingham, most of the operations effected are virtually those of final assembly as regards the sheet-metal components on the motor-cycle, although many of the mechanical components for the engine are manufactured there.

In the main frame of the motor-cycle the petrol tank is fitted on rubber mountings by three-point suspension and thus provision is made for this on the bottom part of the tank. The rear of this portion has two threaded bosses welded to the inside of the tank, locating on two holes pierced in the tank wall. On to these two bosses a right-

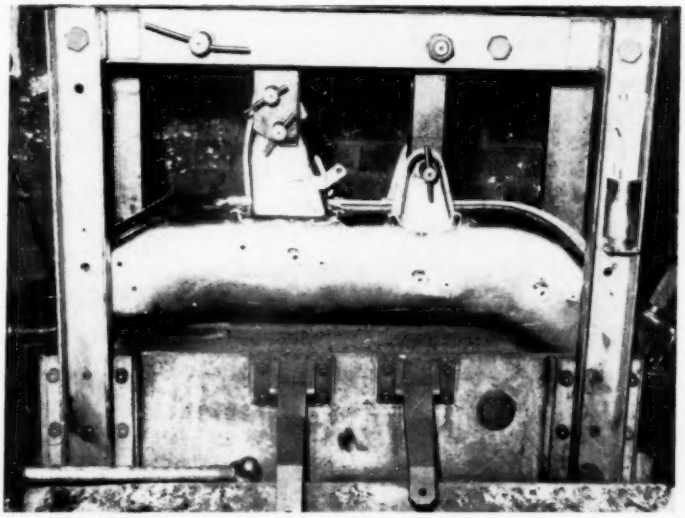
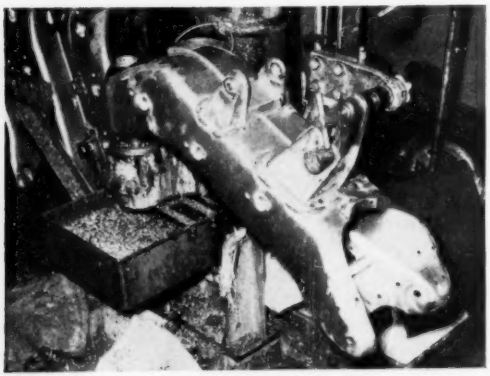


Fig. 37.—*Frame assembly jigged for final welding of engine-mounting brackets, mudguard support bracket and six panel nuts*



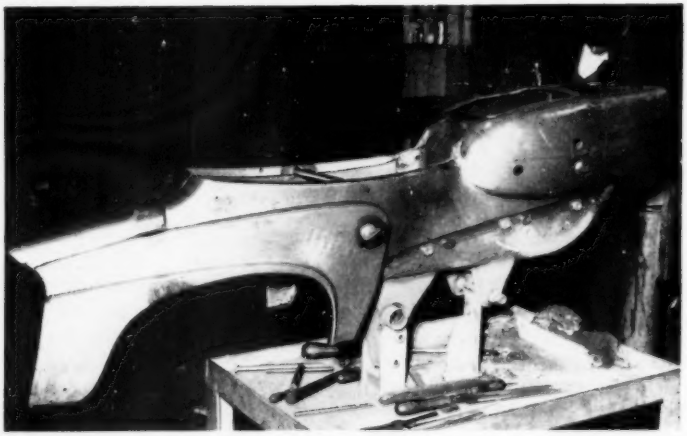
The main frame, also consisting of two halves seam welded together, undergoes several operations before it is ready for use on the motor-cycle. As received, the frame is complete with the rear engine mounting bracket which also incorporates the air silencer, and the front engine mounting bracket. In addition, the lower part of the upswept front portion of the frame has been stiffened internally by the welding in of an extra thickness of metal.

At Selly Oak the frame is first jigged to ensure that the various holes, of which there are about 50, are accurately positioned. The frame is then jigged upside down and the front and rear engine mounting brackets are arc welded (16-gauge rod) (in addition to the spot welds previously made) to the frame

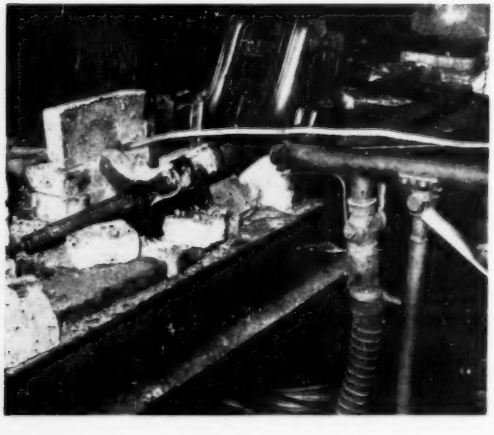
Fig. 38 (*above*).—*Machining steering head recess to take bearing cups. Location is from engine mounting bracket*

Fig. 39 (*right*).—*Tail-cover assembly fitted to body*

Fig. 40 (*below*).—*Brazing steering crown to fork legs*



angled bracket containing four holes is bolted. A similar boss is provided for the third mounting portion, but this is incorporated in a simple formed bracket spotwelded to the forward end of the bottom tank pressing.



for additional strength (Fig. 37). In this jig the rear mudguard support bracket and six threaded bosses on to which the body is attached are also arc welded into position in previously pierced holes.

The next operation is to machine the steering head recesses that take the bearing cups (Fig. 38). To do this the frame is first jigged upside down, locating in the holes in the engine mounting brackets, and then the whole assembly is turned upside down, rejigged, and the other end machined. These two operations are effected on a drilling machine on to which the jigs are attached.

Following this the rear damper-unit mounting, the steering head angle and the engine mounting bracket centres are again gauged and set. Various brackets, such as the horn mounting bracket, are also spot welded on, after which the whole frame is cleaned, Bonderized and coated with grey primer.

The body shell is set on a master frame assembly and final spotwelding carried out, and then the tail end assembly fitted (Fig. 39). This complete assembly is then cleaned, Bonderized and sprayed with grey primer.

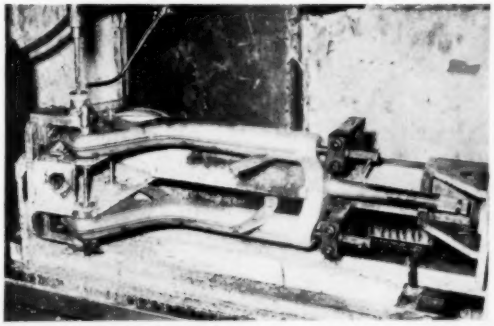


Fig 41 (above).—Drilling projection-welded bosses on fork-legs

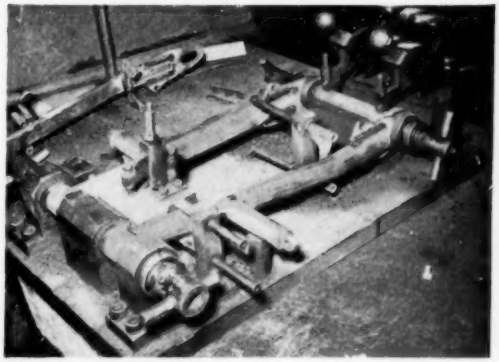


Fig. 42 (above).—Rear-fork components jigged for welding

Fig. 44 (below).—Front forks with front suspension assembled and half of front mudguard

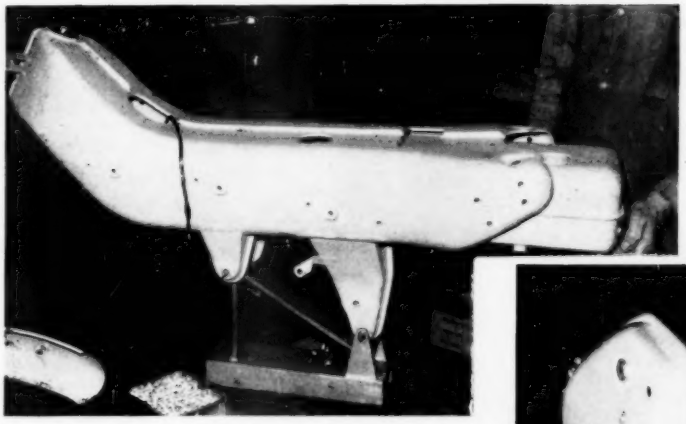


Fig. 43 (above).—Assembling petrol tank into main frame

Side panels are also set on a master pattern and planished where necessary (Fig. 28).

The legs of the front fork assembly arrive at Selly Oak assembled. After checking each leg assembly and removing any surplus metal a pair of legs is brazed on to the malleable cast-iron crown, on to which the steering column is also brazed (Fig. 40). The U-section brackets to which the front and rear parts of the front mudguard are attached, are arc welded to the front fork assembly, one bracket to each leg. The two bosses, projection welded to the lower end of each leg and in which the front suspension pivots, are drilled in a turnover type jig attached to a vertical drilling machine (Fig. 41). The front fork assembly is then Bonderized and painted with primer, after which the front suspension is assembled (Fig. 44).

The two halves of the front mudguard have top-hat sections spot welded inside into which the brackets on the fork legs locate. The mudguards are Bonderized, primed and sprayed with colour. Mudguards are attached to the front fork legs by captive nuts.

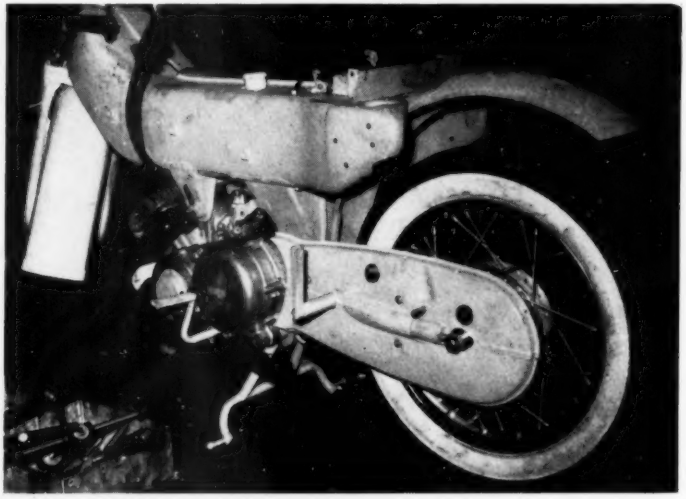


Fig. 45 (right).—Early stage in final assembly sequence (prior to track). The chain case pressings are shown assembled

Fig. 46 (below).—A "Leader" nearing the end of the assembly line

Painting

All components are Bonderized and primed as early as possible, and some components, *e.g.*, the chassis frame and fuel tank, are left in this condition.

The primer paint is of the zinc-chromate type and components are transported after spraying on a Teleflex conveyor through a Parkinson-Cowan infra-red oven in which the stoving temperature is 600° F. for 9 min.

Finish coats are in synthetic paint; they are stoved in a Carrier Engineering camel-back gas-fired convector oven for 1½ hours at 240° F. All finish coats are polished after stoving with car polish.

Spraying is effected in water-wash spray booths of Bullows and Aerospray manufacture; these firms also supplied the spray guns used.

Rear Fork

This is a welded fabrication made from D-section tube. The component parts of the assembly are jigged and arc welded (Fig. 42).

Assembly Sequence

First the electrics, ignition coil, etc., are assembled into the main frame. The fuel tank is fed into the frame from the rear (Fig. 43) and mounted on its three rubber pads. The rear bulkhead is bolted in so that if necessary at any time the fuel tank can be removed. The bulkhead stiffens up the rear end of the frame in conjunction with the rear end bracket which is also bolted in. This bracket, in 9-gauge mild steel, fits under the rear of the frame. A U-section stiffener is spot welded in at the rear of the opening left for the battery, then a further bracket to which the rear mudguard is attached is spot welded on to the underside of the frame.

The battery carrier is dropped into the aperture provided and bolted in; the tool tray is also dropped on and two body brackets underneath the mudguard are fitted in the bottom of two centre holes provided. The frame then goes to the final assembly track after the front forks (together with suspension) and steering column and the rear mudguard have been fitted.

The engine is put on a jig table at the beginning of the track and the frame dropped on to it (Fig. 45);

(*Continued in page* 62)

THE LOCAL BUCKLING OF ALUMINIUM PLATE ELEMENTS

By K. SUTTER*

(*A paper presented at the Symposium on "Applications of Sheet and Strip Metals in Building," organized by the Institute of Sheet Metal Engineering, London, April 29 and 30, 1959.*)

THE economics of the use of aluminium for structural members is based on the possibility of using "blown-up" sections. "Blown-up" sections have large overall dimensions and relatively small wall thicknesses.

The application of the principle of "blowing-up" the section however has limitations. In general, a compression member should not be larger than about a tenth of its length so that the secondary stresses at connexions or nodal points remain negligible. The "blowing-up" is also limited when the space available is restricted. The most severe limitation, however, is due to the fact that the member with a "blown-up" cross-section will fail through local buckling of its components before the member buckles as a whole. In order to design economically the local buckling should occur simultaneously with the general buckling.

"Simultaneously" means here that the compression member fails locally under increasing load just before general buckling would set in. This sequence is important when interpreting the calculations as, if it is not respected, the deformation of the compression member becomes finite instead of very small and the stress distribution over the cross-sections is erroneous if calculated according to the usual procedures of stress analysis.

1. CALCULATION METHODS FOR RECTANGULAR PLATES

The local buckling of plate elements can be dealt with in theory by various methods. These plate elements behave as rectangular plates which are very long with respect to their width. The exact analysis of these plates which compose a section is based on the assumption that they influence each other mutually. According to this method, each type of section must be analyzed separately. With aluminium sections which are supplied in many various shapes, this procedure is not acceptable for the designer.

Noticeable simplification of the calculation is obtained when each plate is considered by itself and the influence of adjacent plates is introduced as a coefficient of restraint. This approximate method will be used hereafter.

1,1. Generally Known Basis of Calculation

The critical buckling stress of rectangular plates which are uniformly compressed in the direction of the longitudinal edges is calculated as follows :—

$$\beta_k = \pi^2 . E . (t/b)^2 . k . /[12(1 - (1/m)^2)]$$

herein E = modulus of elasticity ;
t = thickness of plate ;
b = width of plate ;
$1/m$ = Poisson's constant = 0.33 for aluminium ;
k = coefficient for buckling of rectangular plates.

In order to be in a position to read the critical stresses β_k from a buckling diagram for aluminium alloys, it is necessary to introduce in the above formulae a fictitious slenderness ratio (l_k/i) which has the following definition :—

$$(l_k/i) = \sqrt{12 . (1 - (1/m)^2)/k} . b/t$$

To simplify this expression a new coefficient k' valid specially for local buckling is introduced :—

$$k' = \sqrt{12 (1 - (1/m)^2)/k} = 3.27/\sqrt{k}$$

With this the fictitious slenderness ratio is obtained :—

$$(l_k/i) = k' . b/t$$

and

$$\beta_k = \pi^2 . E/[l_k/i]^2$$

With this transformation of the equation β_k it is now possible to calculate the critical buckling stresses for local buckling with the Euler formula or they can be read with the entry (l_k/i) from a theoretical buckling diagram. When doing so not only is the elastic behaviour taken into account —upon which the Euler formula is based—but also the inelastic behaviour of the material.

The coefficients k or k' introduced in the foregoing depend upon :—

(i) The edge conditions at longitudinal edges ;
(ii) The stress distribution over the width b ;
(iii) The aspect ratio a/b of the rectangular plate, where a = length of the plate.

* Aluminium Laboratories Ltd., Geneva.

TABLE I (para. 1,1,1)—*Coefficients k and k′ for Webs with Identical Coefficients of Restraint at the Edges*

Coefficient of restraint c	k	k′	Half-wave length $d_r.b$
∞ (freely supported) ..	4	1.63	1 . b
5	4.14	1.61	0.968 . b
2	4.35	1.57	0.930 . b
1	4.61	1.52	0.878 . b
0.5	5.01	1.46	0.820 . b
0.2	5.62	1.38	0.755 . b
0.1	6.04	1.33	0.721 . b
0.05	6.35	1.30	0.701 . b
0.02	6.65	1.27	0.687 . b
0 (fixed)	6.97	1.24	0.668 . b

These coefficients can be taken from the known technical literature. They are enumerated in the following to show that the method of dealing with the local buckling of plate elements is based upon known facts.

There is no need to analyze further the influence of the aspect ratio a/b as plate elements in all important cases in practice possess a large aspect ratio and therefore they need not be calculated with the more complicated formulae valid for plates with low aspect ratio.

1,1,1. *THE INFLUENCE OF THE EDGE CONDITIONS*

If the stress distribution is uniform over the width b and the aspect ratio a/b is large, the coefficients k or k' depend only on the edge conditions at longitudinal edges and the coefficients of restraint c. The two basic cases can be distinguished as :—

(i) The plate is supported on both edges and the coefficients of restraint are identical for both edges ; such plates will be termed, in the following, " webs " ;

(ii) The plate is supported only on one edge and on the other edge it is free ; such plates will be called " flanges."

This primary sub-division is completed by introducing various coefficients of restraint c. The limiting values for c are infinite for the simply supported edges and zero for the fixed edges (Encastré). (Tables I and II).

Table I requires some comments. If the coefficients of restraint on the two edges a and b of the web are c_a and c_b, *i.e.*, they are different from each other, an average k-value can be used.

$$k = (k_a + k_b)/2$$

k_a and k_b are the coefficients which correspond to the coefficients of restraint c_a and c_b.

As far as the information given in Table II is concerned, the various sources do not give identical values for k for flanges, since these k values depend upon Poisson's constant and also upon the chosen limiting aspect ratio a/b.

TABLE II (para. 1,1,1)—*Coefficients k and k′ for Flanges where the Half Wave Length is Shorter than a*

Coefficient of restraint c	k according to Bleich	k used by us	$k' = \frac{3{,}27}{\sqrt{k}}$	Half wave length $d_F.b$
∞ (freely supported)	0.425	0.406	5.13	∞ . b
500	0.445	0.425	5.02	9.50 . b
200	0.46	0.44	4.93	7.59 . b
100	0.475	0.455	4.85	6.33 . b
50	0.495	0.475	4.74	5.37 . b
20	0.535	0.515	4.55	4.27 . b
10	0.58	0.56	4.37	3.62 . b
5	0.64	0.615	4.17	3.08 . b
2	0.75	0.725	3.84	2.60 . b
1	0.85	0.83	3.59	2.21 . b
0.5	0.965	0.935	3.38	1.98 . b
0.2	1.095	1.065	3.17	1.78 . b
0.1	1.175	1.145	3.06	1.72 . b
0.05	1.225	1.195	2.99	1.69 . b
0 (fixed) ..	1.277	1.246	2.93	1.668. b

From the column of the half wave-length of the buckles $d_F.b$ it can be inferred that with short flanges the wave-lengths corresponding to high coefficients of restraint cannot occur in practice as the flanges are not infinitely long. In this exceptional case it is permitted, in accordance with the theory of buckling of rectangular plates, to increase the k values or to decrease the k' values in accordance with the following formulae :—

$$k = 0.406 + b^2/a^2$$

and

$$k' = 3.27/\sqrt{.406 + b^2/a^2}$$

a = length of flange ; b = width of flange.

The author does not recommend designers to use the above formulae because, as already mentioned, this case only occurs with high coefficients of restraint. High coefficients of restraint can only be expected in flanges if these are restrained by other flanges. Such sections are characterized by the fact that they possess only one edge on which all the plate elements join, as is the case for instance for angle, T, or cruciform sections.

TABLE III (para 1,1,2)—*Coefficients of k and k′ for Webs with Identical Coefficients of Restraint at the Edges*

stress ratio $s = \sigma_{min}/\sigma_{max}$	edges fixed			edges freely supported		
σ_{min} – $+\sigma_{max}$	k	k′	d_S	k	k′	d_S
– 1	6·97	1·24	0·668 b	4·00	1·63	1·00 b
0·75	7·87	1·17		4·51	1·54	
– 0·5	9·27	1·08	0·660 b	5·32	1·42	0·998 b
0·25	11·16	0·98		6·42	1·29	
0 0	13·54	0·89	0·653 b	7·81	1·17	0·983 b
– 0·25	17·74	0·78		10·00	1·03	
– 0·5 +	23·47	0·68	0·608 b	13·40	0·89	0·915 b
– 0·75	30·71	0·59		18·03	0·77	
– 1 +	39·52	0·52	0·446 b	23·88	0·67	0·667 b

TABLE IV (para 1,1,2)—*Coefficients of k and k' for Webs with Different Coefficients of Restraint at the Edges*

stress ratio $s = \sigma_{min}/\sigma_{max}$		σ_{max} at the fixed edge			σ_{max} at the freely supported edge		
		c = 0	c = ∞		c = ∞	c = 0	
σ_{min} – $+\sigma_{max}$		k	k	ds	k	k	ds
–	1	5·41	1·41	0·800 b	5·41	1·41	0·800 b
	0·75	6·27	1·31		5·97	1·34	
–	0·5	7·62	1·18		6·85	1·25	
	0·25	9·44	1·06		8·04	1·15	
– 0	0	11·73	0·95	0·763 b	9·54	1·06	0·797 b
	– 0·25	15·80	0·82		11·39	0·97	
– +	– 0·5	22·45	0·69		14·41	0·86	
	– 0·75	29·69	0·60		18·59	0·76	
– +	– 1	39·52	0·52	0·446 b	23·94	0·67	0·667 b

TABLE V (para 1,1,2)—*Coefficients of k and k' for Flanges with Maximum Compressive Stress σ_{min} at the Supported Edge*

stress ratio $s = \sigma_{min}/\sigma_{max}$		fixed edge			freely supported edge		
σ_{min} – $+\sigma_{max}$		k	k'	d_f	k	k'	d_f
–	1	1·246	2·93	1·63 b	0·406	5·13	∞
	0·75	1·61	2·58		0·51	4·58	
–	0·5	2·23	2·19		0·68	3·97	
	0·25	3·34	1·79		0·98	3·31	
– 0	0	5·80	1·36	1·58 b	1·63	2·56	∞
	– 0·25	10·70	1·00		3·16	1·84	
– +	– 0·5	19·00	0·75		6·85	1·25	
	– 0·75	30·70	0·59		14·10	0·87	
– +	– 1	39·52	0·52	1 b	23·67	0·67	∞

With such sections the centre of shear coincides with this edge and the warping constant is zero. They are, therefore, prone to fail as a whole in torsional buckling and torsional buckling is in this case more unfavourable than local buckling. The refinement brought about with the above formulae is therefore irrelevant. Incidentally, the torsional buckling referred to above occurs in one-half wave and the notion of coefficient of restraint is of no significance.

1,1,2. *THE INFLUENCE OF STRESS DISTRIBUTION*

The influence of stress distribution upon the k and k' values under given edge conditions and coefficients of restraint can also be taken from the known technical literature. The following cases can be distinguished :—

(i) The plate is supported along two edges and the coefficients of restraint at the longitudinal edges are identical (Table III). This is the case with webs of sections which have symmetrical flanges with respect to the transverse axis of the web.

(ii) The plate is supported along two edges and the coefficients of restraint at the two edges are not identical (Table IV). This is the case with webs of sections with flanges which are not symmetrical with respect to the transverse axis of the web.

TABLE VI (para 1,1,2)—*Coefficients of k and k' for Flanges with Maximum Compressive Stress σ_{min} at the Non-supported Edge*

stress ratio $s = \sigma_{min}/\sigma_{max}$		fixed edge			freely supported edge		
σ_{min} – $+\sigma_{max}$		k	k'	d_f	k	k'	d_f
–	1	1·246	2·93	1·668 b	0·406	5·13	∞
	0·75	1·31	2·86		0·43	4·97	
–	0·5	1·38	2·79		0·46	4·80	
	0·25	1·46	2·71		0·50	4·63	
– 0	0	1·56	2·62	1·667 b	0·54	4·45	∞
	– 0·25	1·69	2·52		0·59	4·24	
– +	– 0·5	1·81	2·43		0·66	4·02	
	– 0·75	1·94	2·35		0·73	3·83	
– +	– 1	2·07	2·27	1·63 b	0·81	3·64	∞

(iii) The plate is supported at one edge only and the maximum compressive stress occurs at this supported edge (Table V). This may be the case with flanges.

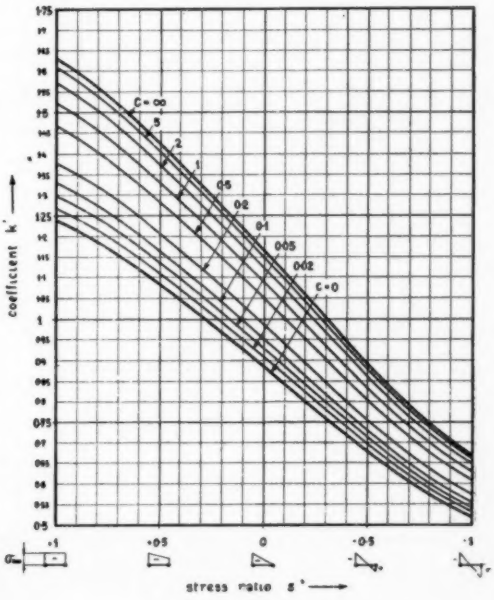
(iv) The plate is supported along one edge only and the maximum compressive stress occurs at the unsupported edge (Table VI). This may be the case with flanges.

In Tables III to VI the stress ratio s varies

Fig. 1

DIAGRAM 1 OF § 121

k'-values for webs with identical coefficients of restraint c at the edges



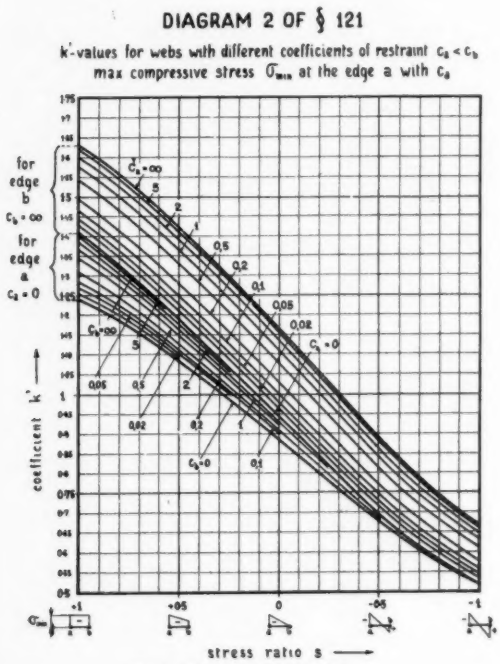


DIAGRAM 2 OF § 121
k'-values for webs with different coefficients of restraint $c_a < c_b$
max compressive stress σ_{max} at the edge a with c_a
for edge b $c_b = \infty$
for edge a $c_a = 0$
coefficient k'
stress ratio s

Fig 2 (*above*). *Fig.* 4 (*below*)

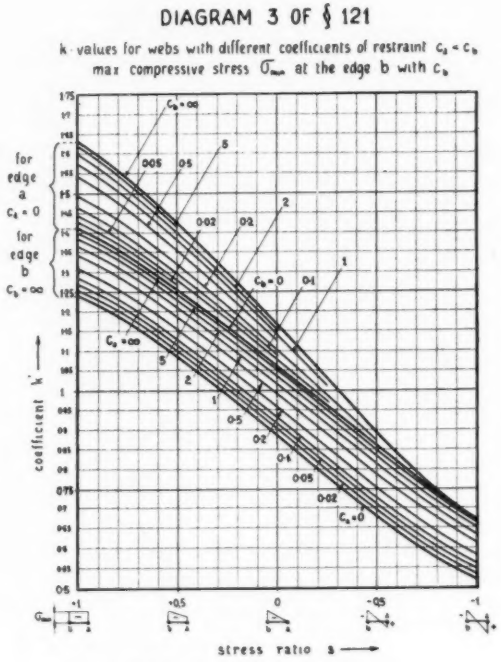


DIAGRAM 3 OF § 121
k-values for webs with different coefficients of restraint $c_a < c_b$
max compressive stress σ_{max} at the edge b with c_b
for edge a $c_a = 0$
for edge b $c_b = \infty$
coefficient k'
stress ratio s

Fig. 3 (*above*). *Fig* 5 (*below*)

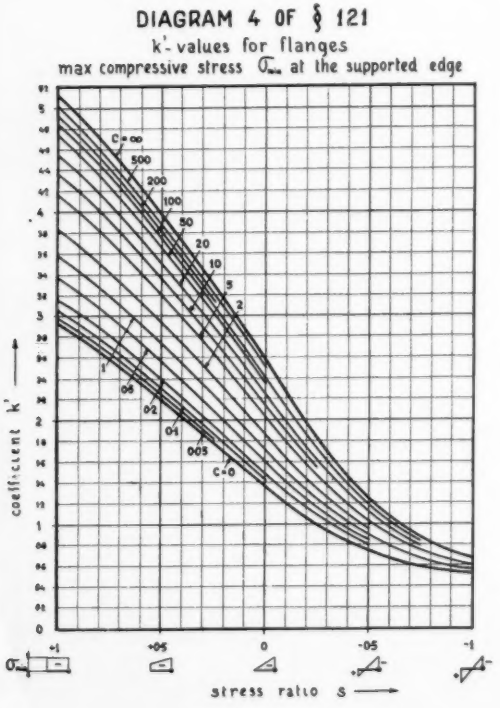


DIAGRAM 4 OF § 121
k'-values for flanges
max compressive stress σ_{max} at the supported edge
coefficient k'
stress ratio s

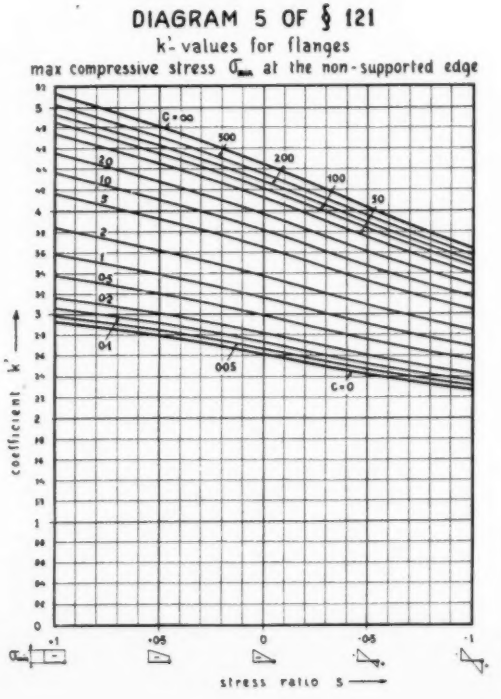


DIAGRAM 5 OF § 121
k'-values for flanges
max compressive stress σ_{max} at the non-supported edge
coefficient k'
stress ratio s

between +1 and −1. Values beyond these limits are not given by the technical literature.

The k values underlined in these tables are taken from technical literature and those not underlined have been determined by the author.

1,2. Practical Procedure for Calculating Local Buckling of Plate Elements

The above summary of known facts permits the development of a method of calculation for the critical local buckling stresses which is satisfactory for the designer. The designer determines by means of the statical calculation of the structure the stress ratio s of the plate element of width b and thickness t. Subsequently he determines in accordance with the detailed indications contained in the following paragraph 2, the coefficient of restraint c or the coefficients of restraint c_a and c_b. He analyzes each plate element by itself and in applying the information given in paragraph 2 he also determines whether this plate element is really buckling.

1,2,1. *DIAGRAMS FOR THE k′ VALUES*

At this juncture he reads from Figs. 1 to 5 the k' values which are related to the quantities c and s. With this k' value he calculates a fictitious slenderness ratio as follows :—

$$(l_k/i) = (b/t) \cdot k'$$

and he uses this fictitious slenderness ratio for the calculation, or for reading the critical buckling stress from a theoretical buckling diagram. The permissible stresses are obtained by dividing the critical buckling stress by the safety factor.

These k' diagrams were established for the following five cases :—

(i) Webs with identical coefficients of restraint on both edges (Fig. 1) ;

(ii) Webs with different coefficients of restraint $c_a < c_b$ and where the maximum compressive stress σ min. occurs at the more severely restrained edge, *i.e.*, at the edge with the smaller coefficient of restraint c_a (Fig. 2) ;

(iii) Webs with different coefficients of restraint $c_a < c_b$ where the maximum compressive stress σ min. occurs on the less severely restrained edge, *i.e.*, at the edge with the bigger coefficient of restraint c_b (Fig. 3) ;

(iv) Flanges where the maximum compressive stress occurs at the supported edge (Fig. 4) ;

(v) Flanges where the maximum compressive stress occurs at the unsupported, *i.e.*, free edge (Fig. 5).

In the k' diagrams a sufficient number of curves for c values are given so that accurate interpolation of the k' values is possible. The entry for the c values is intentionally not made on one of the axes of coordinates. The reason for this will be seen later. The choice of the intervals between the

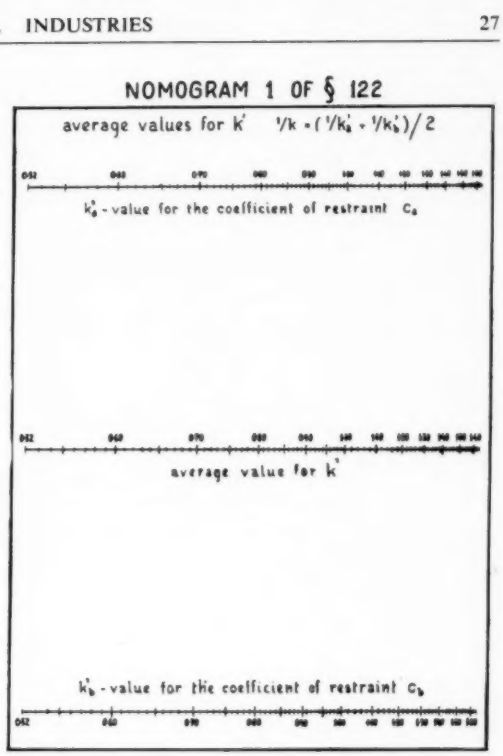

NOMOGRAM 1 OF § 122
average values for k'
1/k = (1/k'a + 1/k'b)/2
k'a - value for the coefficient of restraint Ca
average value for k'
k'b - value for the coefficient of restraint Cb

Fig. 6

c curves which follow the progression 1, 2, 5, 10, 20, 50 and so on, is dictated by reasons of legibility.

The two cases according to Table IV (para. 1,1,2) appear in Figs. 2 and 3 of paragraph 1,2,1, as curves for $c_a = 0$ and $c_b = \infty$ or $c_a = \infty$ and $c_b = 0$. These two diagrams are so arranged that determining averages for the k' values is also possible with unequal stress distribution.

1,2,2. *RULES FOR THE USE OF THE k′ DIAGRAMS*

The use of the diagrams in Figs. 1, 4 and 5 (para. 1,2,1) does not offer any difficulty. With Figs. 2 and 3, however, the following must be taken into account. The two coefficients of restraint c_a and c_b at the two edges are different. In order to obtain a definite k' value, an average must be calculated. One therefore reads with the entries s and c_a a first value $k' = k_a'$ and with the entries s and c_b a second value $k' = k_b'$. Then the determinant average is calculated according to the following formula :—

$$1/k' = (1/k_a' + 1/k_b')/2$$

As it is usual in the theory of plate buckling to use the averages—

$$k = (k_a + k_b)/2$$

(*see* para. 1,1,1) it is necessary to take their reciprocals in the case of the average of the k' values. The use of the nomogram (Fig. 6) simplifies the calculation of these averages.

The above procedure means that the plate with the coefficients of restraint c_a and c_b is calculated as follows: first as a plate with a given value c_a and $c_b = \infty$ and second, as a plate with the given c_b and $c_a = 0$. The k' values thus obtained are used to calculate an average value.

2. Discussion of the Variables Appearing in the k′ Diagrams

The five diagrams for k' can be used when the width b and the thicknesses t of the individual plate elements are constant. This is mostly the case with integral sections which are obtained for instance by brake pressing from sheet material or welding up of sheet material. This may also be the case with extrusions. How built-up sections which are obtained by riveting together individual sections are calculated, will be shown later in para. 2,2. It will also be shown later, in para. 2,1,3, how sections are calculated where the widths b and thicknesses t are variable.

2,1. Integral Sections

When the above-mentioned k' diagrams are applied to integral sections, the only variables which appear are the stress ratio s and the coefficient of restraint c. These will be investigated more thoroughly hereafter.

2,1,1. *THE STRESS RATIO s*

There is not much need to discuss the stress ratio s. The distribution of stresses over the investigated plate element is known from the static calculation. It is permissible to take into account only the normal stresses and to neglect shear stresses. This stress ratio is plotted on the axis of the abcissae and the k' value sought is the corresponding ordinate which is cut off by the curve of the determinant coefficient of restraint c.

The stress ratio s is given within the limits of $+1$ and -1. Values smaller than -1 are not known. But, as with these values tensile stresses are predominant, the plate element is not likely to fail by local buckling.

With the above the influence of s, however, is not fully described. As will be seen later, the stress ratio s influences also the coefficients of restraint and in consideration of this fact the two cases will be distinguished:—

(i) The restrained and the restraining plate elements have the stress ratio $s = 1$;

(ii) The restrained and the restraining plate elements have a stress ratio which is different from 1.

2,1,2. *COEFFICIENTS OF RESTRAINT c*

The coefficients of restraint c may be estimated by the experienced designer; however, errors are avoided if these coefficients of restraint are calculated. The method outlined by F. Bleich in his book "Buckling Strength of Metal Structures" (1952) will therefore be applied.

The coefficients of restraint c are calculated with a formula of the following type:—

$$c = (t_1/t_2)^3 \cdot d \cdot f\{b_2/(d \cdot b_1)\} / \{1 - g \cdot (t_1/t_2)^2/(b_1/b_2)^2\}$$

where t = thickness of plates;
b = width of plates;
d = half wave-length of the buckles divided by the width of the plate which buckles;
$f\{b_2/(d \cdot b_1)\}$ is a function which takes into account the ratio of widths of the two plates;
g = coefficient depending upon the stress ratios s_1 and s_2 of the two plates.

Index 1 refers to the restrained plate and Index 2 to the restraining plate. In the sketches referred to later the restrained plate will be shown in black and the restraining plate in white. With sections with more than two plates, a co-restrained plate is shaded and marked with the index 0. A co-restraining plate is shown in white and receives Index 3.

When using the notion of coefficient of restraint it is important to remember that the coefficient of restraint c of plate 1 produced by plate 2 is not the coefficient of restraint of plate 2 produced by plate 1. The interaction of the two plates is such that the angle of rotation of the restrained (and buckling) plate and the restraining plate with respect to their common edge are equal. The restraining plate remains straight if it is a flange and it will be curved if it is a web, as the second edge of a web is not assumed to alter its position.

TABLE VII (para. 2,1,2)—*Coefficient g for Uniform Stress Distribution*

Stress ratio s	Restrained plate element 1		Restraining plate element 2		Coefficient g
	Support	k_1	Support	k_2	
1	One edge	0.406	One edge	0.406	1
1	One edge	0.406	Both edges	4	0.1015
1	Both edges	4	One edge	0.406	9.852
1	Both edges	4	Both edges	4	1

From these angles of rotation, bending moments with respect to the common edge are calculated in the plates and the factor of proportionality between angle of rotation and bending moment is the restraint and not the coefficient of restraint c. The coefficients of restraint c are inversely proportional to the restraint but they are not their reciprocal values.

In order to clarify the procedure for the systematical calculation of the c values, it is necessary to investigate the denominator of the formula. This denominator should be zero and consequently $c = \infty$ if the restrained and the restraining plate buckle simultaneously as simply supported plates. This is the case when :—

$$k_1 (t_1/b_1)^2 = k_2 (t_2/b_2)^2 \text{ or when—}$$
$$g = k_1/k_2$$

The coefficients k_1 and k_2 have been given in para. 1,1,1 and para. 1,1,2. For the various combinations of plates and the stress ratio $s = 1$ the values for g given in Table VII are obtained :—

If in future, therefore, two different cases are distinguished with respect to the stress ratio, it is possible to say roughly that the first case refers to uniformly distributed compression and the second refers to non-uniformly distributed compression. Incidentally some of the coefficients g for the second case are already contained in the first case.

2,1,2.1. *Uniform Stress Distribution*

To obtain a procedure of calculation which is acceptable to the designer, it is necessary to subdivide the important case of uniform stress distribution. This sub-division will be made according to types of sections. Standard sections will be investigated and diagrams will be established of the c values from which the k' values follow directly. Non-symmetrical standard sections will also be investigated where the c or k' values are obtained as averages. These are sections for which, on account of lack of symmetry, only approximate calculation is possible.

Bulb sections will not be dealt with separately in this study. Bulbs are considered as flanges and the resulting combinations of plates are to be allotted to one of the described types. This reference is only to the usual rectangular or trapezoidal bulbs which have small widths b and great thicknesses t, and which are, as a rule, restraining plate elements. Should, exceptionally, circular bulbs be investigated they must be replaced by geometrically equivalent rectangular bulbs.

Finally, some information will be given about special sections which cannot be considered as unsymmetrical standard sections. As can be seen from the above, the sub-division of the sections is made from a purely practical point of view.

2,1,2.1,1. *STANDARD SECTIONS.* In order to obtain a classification with which the designer is familiar, the group of standard sections can be sub-divided into four different types.

Type 1 consists of flanges. The basic type is the angle section.

Type 2 consists of flanges and a web, the flanges are the restrained plate elements. The basic type is the flanges of U sections.

Type 3 consists of flanges and a web, the web is the restrained plate element. The basic type is the web of U sections.

Type 4 consists solely of webs. The basic type is the rectangular tube.

In the c diagrams the ratio of thicknesses and the ratio of widths of the plate elements are the axes of coordinates. The ratio of thickness should lie within the limits of 0.4 and 2.5. Sections with a ratio of thickness t_1/t_2 which is not within these limits are not to be recommended.

The limits of the ratio of widths are different for the four above-mentioned basic types of standard sections. Also in this case no section should be used which has a ratio of widths outside of the given limits. These limits have been established for design reasons.

It is not accidental that in the case of angles the ratio of widths b_1/b_2 varies between 0.4 and 2.5, for flanges of U sections between 0.1 and 1.25, for webs of U sections between 0.8 and 10 and for rectangular tubes between 0.2 and 5.

2,1,2.1,1,1. *Angle sections.* With $g = 1$ the formula for the coefficient of restraint c of an angle section now reads as follows :—

$$c = (t_1/t_2)^3 \cdot d_F \cdot f_F \{b_2/(d_F \cdot b_1)\} / \{1 - (t_1/t_2)^2/(b_1/b_2)^2\}$$

The function f_F of $\{b_2/(d_F \cdot b_1)\}$ refers to the restraining flange. It is a combination of hyperbolic functions and awkward to handle. In order to liberate the designer from this calculation, the present series of c diagrams has been established (Fig. 7). The half wave-length $d_F \cdot b_1$ which appears in the above function and which also appears in the coefficient d_F naturally refers to the restrained *i.e.*, the buckling flange. The values f_F and d_F can be calculated according to the book by F. Bleich already mentioned. For doubly symmetrical cruciform sections, the same c values apply as for angle sections.

The value d_F is influenced by the coefficient of restraint c as can be seen from Table II. In order to reduce the number of c diagrams it is necessary to take into account the fact that when doubling or halving the value of c, the wave-length of the buckles only changes very little and can, therefore, be considered in the above given formula as a constant. With this simplification, the c values for the stem of T sections will be $c/2$ and for the flanges of T sections $2c$. These T sections are here taken to be symmetrical.

The justification of this procedure is as follows :

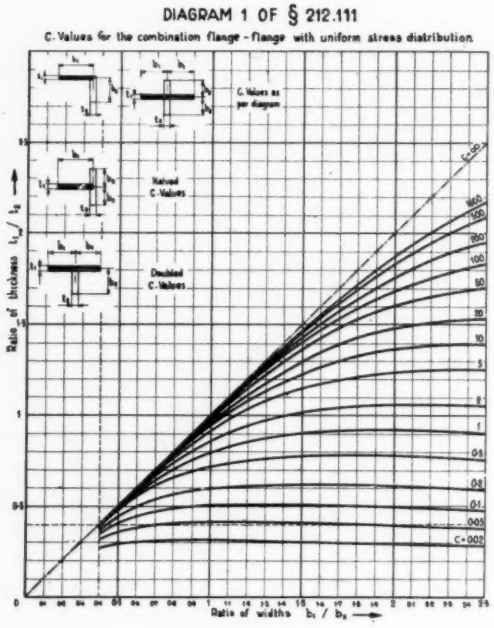


Fig. 7

The stem of a T section with the dimensions b_1 and t_1 is restrained by two identical half flanges of the dimensions b_2 and t_2. The restraint is, therefore, twice as strong as in the case of an angle section for which the coefficient of restraint has been calculated above. As the coefficients of restraint are inversely proportional to the restraints the c value becomes $c/2$.

The half flange of a T section with the dimensions b_1 and t_1 is restrained by the stem of the dimensions b_2 and t_2. The other half flange is co-restrained simultaneously. The restraining effect produced by the stem, therefore, must be divided among two half flanges. The restraining effect, therefore, is only half as big as with the angle section. From this it is inferred that the corresponding coefficient of restraint equals $2c$.

The use of Fig. 7 does not offer any difficulty provided that the given designations are applied systematically. With equal angles the coefficient of restraint is $c = \infty$. The two legs do not influence each other. With the unequal angle, with equal leg thicknesses, the longer of the legs is the restrained flange. If the thicknesses are different it is assumed that one leg is restrained and the other is restraining; if it is not possible to find values for c in the diagram with the entries b_1/b_2 and t_1/t_2, this means that the assumption that one of the two legs is restrained was not correct. The values b_1 and b_2 or t_1 and t_2 therefore must be interchanged.

If Fig. 7 is used for determining the c values of T sections, the procedure is similar. It is assumed for instance that the stem is the restrained plate element; if no c value can be found in the diagram, this means that the half flanges are the restrained plate elements.

The c values for all the types of section referred to above or which can be related to the basic type, are introduced in Figs. 4 or 5 in order to obtain the corresponding k' values. Only those two k' diagrams are used as in all these cases the plate element which buckles is a flange.

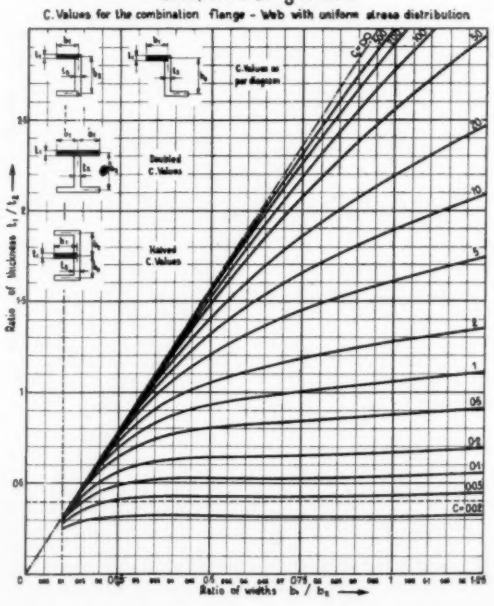
2,1,2.1,1,2. *Flanges of U sections.* With $g = 0.1015$ the coefficient of restraint of the flange of a U section is calculated by means of the following formula :—

$$c = (t_1/t_2)^3 \cdot d_F \cdot f_s \{b_2/(d_F b_1)\} / (1 - 0.1015\,(t_1/t_2)^2/(b_1/b_2)^2)$$

The function f_s of $\{b_2/(d_F \cdot b_1)\}$ refers to the restraining web. It is built-up similarly to function f_F used in the preceding paragraph. The value d_F refers to the restrained flange. For flanges of Z sections the coefficients of restraint are identical. If the same simplifications as already described in para. 2,1,2.1,1,1 are used the coefficients of restraint of flanges of symmetrical H sections are $2c$. For flanges which are restrained by two webs, the corresponding c values are $c/2$ (*see* Fig. 8).

The same rules for the use of this diagram apply as already given under the heading of angle sections.

Fig. 8



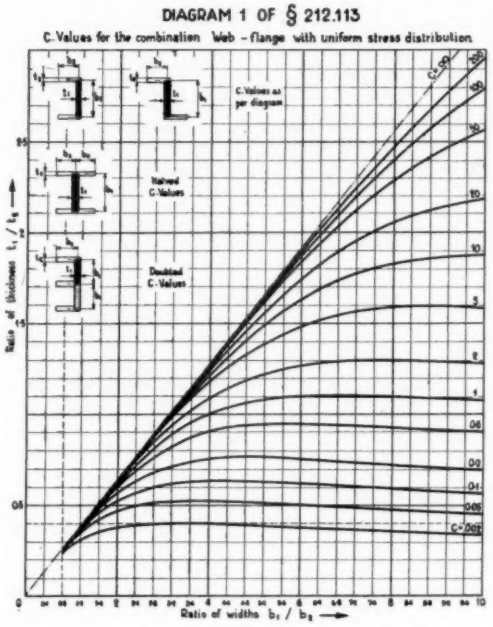


DIAGRAM 1 OF § 212.113
C-Values for the combination Web - flange with uniform stress distribution
C-Values as per diagram
Halved C-Values
Doubled C-Values
Ratio of thickness t_1/t_2
Ratio of widths b_1/b_2

Fig. 9

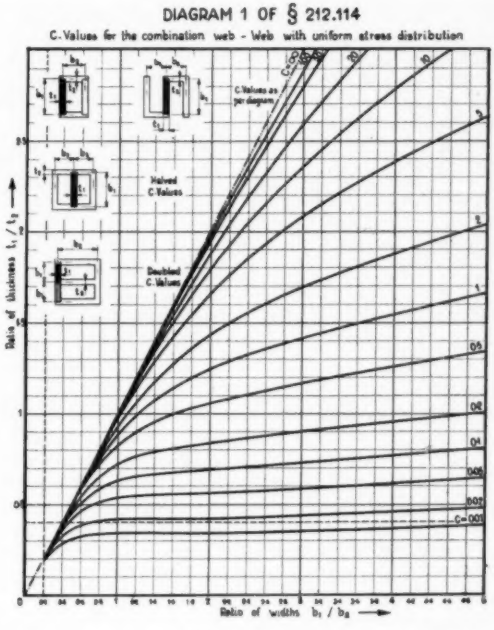


DIAGRAM 1 OF § 212.114
C-Values for the combination web - Web with uniform stress distribution
C-Values as per diagram
Halved C-Values
Doubled C-Values
Ratio of thickness t_1/t_2
Ratio of widths b_1/b_2

Fig. 10

With U or Z sections the first assumption is that the flange is the restrained plate element and that the web is the restraining plate element. If no c values can be found in the diagram for the ratios of widths b_1/b_2 and the ratios of thicknesses t_1/t_2, this means that the flange is the restraining plate element and that, therefore, Fig. 9 must be used. Symmetrical H sections are dealt with in a similar manner.

The c values obtained are introduced in diagram 4 or 5 of para. 1,2,1 in order to obtain the k' values, as also in this paragraph the buckling plate elements are flanges.

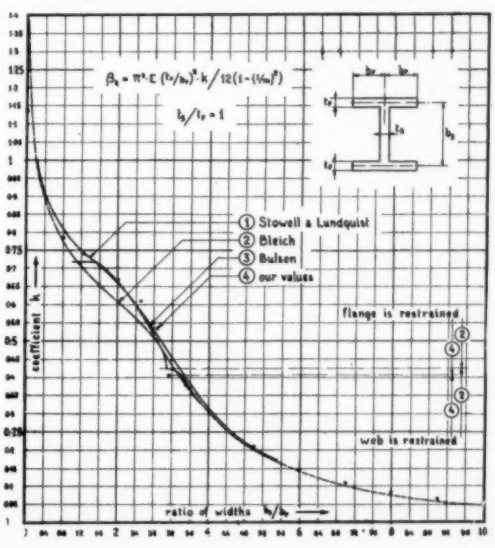
2,1,2.1,1,3. *Webs of U sections.* With $g = 9.852$ the coefficient of restraint of the web of a U section is calculated by means of the following formula :—

$$c = (t_1/t_2)^3 \cdot d_S \cdot f_F \{b_2/(d_S \cdot b_1)\} / (1 - 9.852\ (t_1/t_2)^2/(b_1/b_2)^2)$$

The value d_S refers to the restrained web and the function f_F refers to the restraining flange. Identical c values apply for the webs of Z sections. With the same simplifications as given already in para. 2,1,2.1,1,1 the c values of webs for symmetrical H sections equal $c/2$. If two webs are restrained by the same flange the c values amount to $2c$ (*see* Fig. 9).

The use of this diagram follows the same pattern as already given. As the flanges of the U or Z sections have already been investigated according to the procedure given in para. 2,1,2.1,1,2 there is

Fig. 11



no doubt which one of the plate elements is restrained. The same is true for the other sections given in the sketches in Figs. 8 and 9. The c values so obtained are introduced in Fig. 1 as the above diagram refers to webs with identical coefficients of restraint at their two edges.

2,1,2.1,1,4. *Rectangular tubes.* With $g = 1$ the coefficient of restraint of the webs of rectangular tubes is calculated as follows:—

$$c = (t_1/t_2)^3 \cdot d_s \cdot f_s \{b_1/(d_s \cdot b_1)\} / (1 - (t_1/t_2)^2/(b_1/b_2)^2)$$

The value d_s refers to the restrained webs and the function f_s refers to the restraining webs. With the same simplifications as outlined in para. 2,1,2.1,1,1 the c value of intermediate webs becomes $c/2$. Webs which are only restrained by intermediate webs have a c value of $2c$ (*see* Fig. 10).

The use of this diagram follows a similar pattern to that already described. If the wall thickness of the rectangular tube is constant the longer web is the restrained web. In the case of rectangular tubes with different wall thicknesses, it is necessary to determine by the method of trial and error already described which plate element is restrained. With square tubes with constant wall thickness the coefficient of restraint $c = \infty$.

With the other sections given in the sketches in Fig. 10 the procedure is similar. However, the symmetry given in these sketches must be respected. The c values so obtained are introduced into Fig. 1 (para. 1,2,1) as they refer here to webs with equal coefficient of restraint at their edges.

2,1,2.1,1,5. *Rules for the use of the c diagrams.* When using the c diagrams it will immediately be noticed that the c curves in the neighbourhood of the $c = \infty$ are very close, in fact, so close to each other that they cannot be drawn separately. In this region it is necessary to read the value $c = \infty$. The appearance of the diagram draws the designer's attention to the fact that here the calculation gives doubtful results.

Also the interpolation between the various c curves offers some difficulty. For the normal use of the diagrams it is permissible to indicate the interpolated values as fractions of the distance between the curves. One, therefore, reads for instance $c = 0.5 + 3/10$ths of interval to 1. This is the reason why in the k' diagrams the c values are drawn as curves and not as a scale on one of the axes of coordinates.

If a section other than the basic section indicated on the diagrams is investigated it is satisfactory to use for the double values :

0.02	instead of	0.04	approximately	0.05
0.2	,,	0.4	,,	0.5
2.0	,,	4.0	,,	5.0

In the case of half values it is sufficient to take for the halved values :

0.05	instead of	0.025	approximately	0.02
0.5	,,	0.25	,,	0.2
5.0	,,	2.5	,,	2.0

This liberty accelerates the use of these diagrams without introducing appreciable errors. Had the progression of the c values been chosen according to the series 1, 2, 4, 8, 16, 32, etc., no errors at all would occur, but the diagram would be less easy to read.

The c values indicated in Figs. 7, 8, 9 and 10 which are not contained in the diagrams for the k' values, *i.e.*, Figs. 1 to 5 (para. 1,2,1) serve for the accurate determination of the above-mentioned double or half c values. There is no need to take them into consideration when reading the k' values.

In this paragraph no complete summary of all the possible types of sections has been given for which the c values are either half or double the values given in the diagrams. However, the information given is sufficient to illustrate the principle and the designer will find his way himself, if he has to study one of the types of sections which is only used exceptionally and which is not mentioned.

2,1,2.1,1,6. *Comparison with other methods.* It is desirable to compare at this juncture the k' values obtained by the author's method with those given in the technical literature. Fig. 11 gives this comparison. However, this diagram is not arranged as are Figs. 7 to 10. It is arranged in the way usually used and which needs discussion here in order to make the comparison understandable.

Fig. 11 gives the relation between the ratio of widths and the ratio of thicknesses with the k values directly, that is to say, without passing through the coefficients of restraint c. This is only an advantage if standard sections are investigated and if diagrams of each possible shape have been established. For unsymmetrical standard sections or special sections this means, however, such an amount of work that no designer can afford to undertake it. Introducing the coefficients of restraint c permits the establishment of four basic diagrams for the four combinations of plate elements :—

flange/flange
flange/web
web/flange
web/web

(The buckling plate is shown in italics). With these four basic diagrams the designer can study all the sections without special effort.

The coefficients in Fig. 11 are k values as usual in the theory of plate buckling and not the k' values as used by the author. This is a drawback if the critical buckling stresses are calculated with fictitious slenderness ratios. The apparent complication of the method however is necessary if the non-elastic behaviour of the material is to be taken into account.

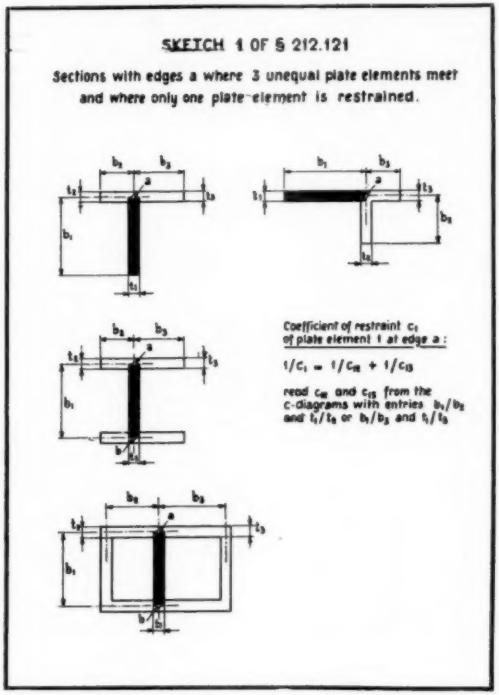

SKETCH 1 OF § 212.121
Sections with edges a where 3 unequal plate elements meet and where only one plate element is restrained.
Coefficient of restraint c_1 of plate element 1 at edge a:
$1/c_1 = 1/c_{12} + 1/c_{13}$
read c_{12} and c_{13} from the c-diagrams with entries b_2/b_1 and t_2/t_1 or b_3/b_1 and t_3/t_1

Fig. 12

Fig. 11 does not apply to an individual plate element like a flange or a web but applies to a complete section, in this case to an H section with symmetrical flanges. Irrespective of which plate element is buckling the entry into the diagram must be made with the ratio of widths b_S/b_F. (b_S means here the width of the web and b_F the width of the flange). When calculating the fictitious slenderness ratio with a large value of b_S/b_F and consequently the *web* is buckling—it is still necessary to use the ratio b_F/t_F of the *flange*. Such a procedure is so illogical that, according to the author's experience, the designers miss the point and introduce erroneously the ratio b_S/t_S instead of b_K/t_K in the formula for the fictitious slenderness ratio.

Therefore in order to compare k' values with the k values used in other methods it is necessary to transform them in such a way that instead of the ratio b_F/t_F the ratio b_S/t_S may be used in the formula for the fictitious slenderness ratio.

This comparison covers the following methods:

(1) Stowell, E. Z. and Lundquist, E. E., Local instability of columns with H, Z, U and rectangular tube sections. NACA Techn. Note 743, 1939.

(2) Bleich, F., Buckling strength of metal structures, McGraw Hill Book Co. Inc., New York, 1951.

(3) Bulson, P. S., Local instability problems of light alloy struts, A.D.A. Research Report No. 29, London, 1955. (This paper takes into account the interaction of all the plate elements present in a section.)

(4) Aluminium Laboratories values.

The values according to 4 match those according to 1 very well with the exception of the transition zone where the buckling of the flanges is superseded by the buckling of the webs. As will be shown later this region gives doubtful results with any method on account of the manufacturing tolerances in the widths and thicknesses of the plate elements.

2,1,2.1,2. *Unsymmetrical Standard Sections.* The four diagrams given in Figs. 7 to 10 are the core of the author's method. They permit not only the determination of the c values for the four basic types of section and for simple shapes which can be derived from them but also the calculations of the c values for unsymmetrical standard sections.

Such unsymmetrical standard sections are for instance T sections with unequal half flanges, H sections with unequal half flanges, that is to say, sections with edges where three plate elements meet which have not identical dimensions and which may be either flanges or webs. Such sections are calculated in two different ways depending whether one or two of the three plate elements which meet at an edge are restrained. The special case where all the three plate elements buckle simultaneously does not offer any difficulty in calculation. None of the three plate elements restrain the two others and, therefore, the c values are ∞.

U, Z and H sections having flanges of unequal dimensions, *i.e.*, which comprise webs of different coefficients of restraint at the two edges, are also unsymmetrical sections. These different types of unsymmetrical standard sections will be analyzed later in more detail.

2,1,2.1,2,1. *Sections with edges a where three unequal plate elements meet and where only one plate element is restrained.* Examples of this type of unsymmetrical section (Fig. 12) are the stems of T sections with unequal half flanges, the web of an H section with unequal half flanges and corresponding variations of the sections shown in the four diagrams Figs. 7 to 10.

The restrained plate 1 has the dimensions b_1 and t_1, the two restraining plates 2 and 3 have the dimensions b_2 and t_2 and b_3 and t_3. The combination of the plates 1 and 2 gives the coefficient of restraint c_{12}. This coefficient can be read from one of the c diagrams of para. 2,1,2.1,1 (Figs. 7 to 10). The plates 1 or 2 may be flanges or webs. The combination of plate 1 and 3 gives the coefficient of restraint c_{13} which is determined in a similar way to c_{12}. The restraints from plate 2 and plant 3 are superimposed, therefore, the coeffi-

cients of restraint c must be calculated in accordance with the following formula :—

$$1/c_1 = 1/c_{12} + 1/c_{13}$$

If, incidentally, $c_{12} = c_{13}$, $1/c_1 = 2/c_{12}$ or $c_1 = c_{12}/2$, this is the special case already indicated in the c diagrams.

If no values can be read in the c diagrams for the ratio of widths or the ratio of thicknesses to be investigated, this means that the assumption which of the three plate elements is the restrained one, was not correct. This assumption, therefore, must be changed until a c value can be read from the diagrams.

As an example for the above the unsymmetrical T section will be calculated according to Fig. 12.

EXAMPLE 1 OF PARA. 2,1,2.1,2,1

Unsymmetrical T-section ; one plate element restrained

Restrained plate element 1 : (flange)
$b_1 = 20$ cm. $t_1 = 2$ cm.
restraining plate element 2 : (flange)
$b_2 = 10$ cm. $t_2 = 2$ cm.
restraining plate element 3 : (flange)
$b_3 = 15$ cm. $t_3 = 3$ cm.
with $b_1/b_2 = 2$ and $t_1/t_2 = 1$ follows from Fig. 7
$c_{12} = 1.6$
with $b_1/b_3 = 1.33$ and $t_1/t_3 = 0.67$ follows from Fig. 7
$c_{13} = 0.25$
thus $1/c_1 = 1/1.6 + 1/0.25 = 4.62$
$c_1 = 1/4.62 = 0.22$
with $c_1 = 0.22$ and $s = 1$ follows from Fig. 4 or 5
$k' = 3.19$
and the fictitious slenderness ratio is :
$(l_k/i) = k' (b_1/t_1) = 3.19 \times 20/2 = 31.9$

The unsymmetrical T section will be calculated as if it were :

(i) A symmetrical T section with two equal restraining half flanges b_2 and t_2, and

(ii) A T section with two equal restraining half flanges b_3 and t_3.

EXAMPLE 2 OF PARA. 2,1,2.1,2,1

Unsymmetrical T-section, one plate element restrained

I.—*Calculation as symmetrical T-section*
restrained plate element 1 : (flange)
$b_1 = 20$ cm. $t_1 = 2$ cm.
restraining plate element 2 : (flange)
$b_2 = 10$ cm. $t_2 = 2$ cm.
co-restraining plate element 3 : (flange)
$b_2 = 10$ cm. $t_2 = 2$ cm.
with $b_1/b_2 = 2$ and $t_1/t_2 = 1$ follows from Fig. 7
$C_{12} = 0.5 \times 1.6 = 0.8$
with $c_{12} = 0.8$ and $s = 1$ follows from Figs. 4 or 5
$k'_I = 3.52$

II.—*Calculation as symmetrical T-section*
restrained plate element 1 : (flange)
$b_1 = 20$ cm. $t_1 = 2$ cm.

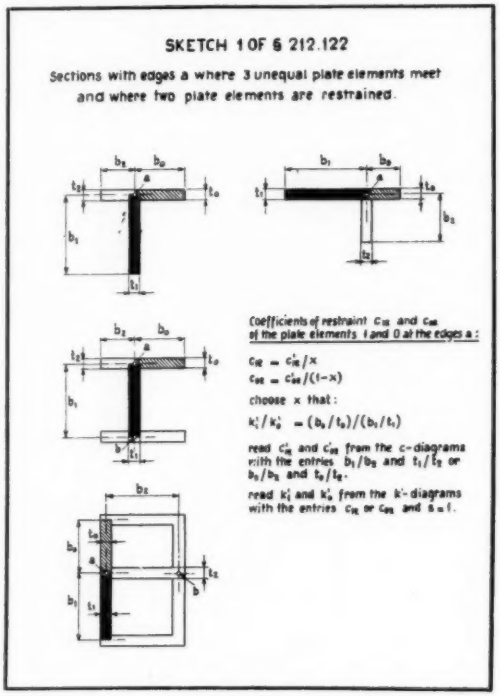

SKETCH 1 OF § 212.122
Sections with edges a where 3 unequal plate elements meet and where two plate elements are restrained.
Coefficients of restraint c_{1a} and c_{2a} of the plate elements 1 and 0 at the edges a :
$c_{1a} = c'_{1a}/x$
$c_{2a} = c'_{2a}/(1-x)$
choose x that :
$k'_1/k'_0 = (b_1/t_1)/(b_2/t_2)$
read c'_{1a} and c'_{2a} from the c-diagrams with the entries b_1/b_2 and t_1/t_2 or b_2/b_1 and t_2/t_1.
read k'_1 and k'_0 from the k'-diagrams with the entries c_{1a} or c_{2a} and s = 1.

Fig. 13

restraining plate element 2 : (flange)
$b_3 = 15$ cm. $t_3 = 3$ cm.
co-restraining plate element 3 : (flange)
$b_3 = 15$ cm. $t_3 = 3$ cm.
with $b_1/b_3 = 1.33$ and $t_1/t_3 = 0.67$ follows from Fig. 7
$c_{13} = 0.5 \times 0.25 = 0.125$
with $c_{13} = 0.125$ and $s = 1$ follows from Figs. 4 or 5
$k'_{II} = 3.08$

III.—*Average*
$k' = (k'_I + k'_{II})/2 = (3.52 + 3.08)/2 = 3.30$
thus the fictitious slenderness ratio is :
$(l_k/i) = k' \cdot (b_1/t_1) = 3.30 \times 20/2 = 33.0$

The result is not the same as in Example 1 of para. 2,1,2.1,2,1 ; this way of calculating the average is only permissible if the k'-values are only slightly different.

2,1,2.1,2,2. *Sections with edges a where three unequal plate elements meet and where two plate elements are restrained.* The same examples as given in Fig. 12 apply. The only difference is that the dimensions of b and t are changed and for this reason two plate elements instead of one are restrained (Fig. 13).

If the restrained plate 1 were present its co-efficient of restraint according to Fig. 7 would be

c_{12}'. In the case where plate 0 alone is present, the coefficient of restraint according to Fig. 7 would be c_{02}'. These values cannot be attained as only one restraining plate element is present. The real coefficients of restraint are, therefore, bigger. Therefore :

$$c_{12} = 1/x \,.\, c_{12}'$$
$$c_{02} = 1/y \,.\, c_{02}'$$

As has already been said, c_{12} is bigger than c_{12}' and c_{02} is bigger than c_{02}' therefore x and y must be smaller than unity.

For the case where the two plates 1 and 0 have identical dimensions—this case has already been indicated on the c diagrams—then : $c_{12} = 2 \,.\, c_{12}'$; $c_{02} = 2 \,.\, c_{02}'$ and $c_{12} = c_{02}$.

Therefore, $x = y = 1/2$ or $x + y = 1$.

For the case where plate 0 is missing $c_{12} = c_{12}'$. x becomes unity and $y = 0$. From this it follows that $c_{02} = c_{02}'/0 = \infty$. No restraint is present as plate 0 is missing.

The equations for the coefficients of restraint can therefore be written as follows :—

$$c_{12} = c_{12}'/x$$
$$c_{02} = c_{02}'/(1\text{-}x)$$

The unknown x is obtained by the following reasoning. As both plate elements 1 and 0 buckle simultaneously their critical buckling stresses and hence their fictitious slenderness ratios must be identical.

$$l_k/i = (b_1/t_1) \,.\, k_1' = (b_0/t_0)\, k_0'$$

From this it follows that :—

$$k_1'/k_0' = (b_0/t_0)/(b_1/t_1)$$

Between the k_1' and the k_0' values and the coefficients of restraint c exists a relation which is given by the k' diagrams. This relation can be expressed by formulae but the solution of the equations is not easy. The method of trial and error is therefore used which is much more rapid. An x value is assumed, with this x value and the known quantities c_{12}' and c_{02}', c_{12} and c_{02} are calculated. Subsequently the corresponding k' values are taken from the k' diagrams and a check is made whether the above condition for the ratio k_1'/k_0' is fulfilled.

A calculated example will show the procedure for the unsymmetrical T section of Fig. 13.

EXAMPLE 1 OF PARA. 2,1,2.1,2,2

Unsymmetrical T-section ; two plate elements restrained

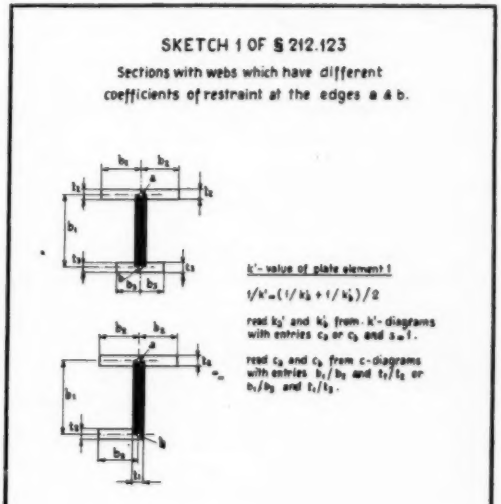
Restrained plate element 1 : (flange)
$b_1 = 20$ cm. $t_1 = 2$ cm.
co-restrained plate element 0 : (flange)
$b_0 = 15$ cm. $t_0 = 1$ cm.
restraining plate element 2 : (flange)
$b_2 = 10$ cm. $t_2 = 2$ cm.
thus $k'_1/k'_0 = (b_0/t_0)/(b_1/t_1) = 1.5$
with $b_1/b_2 = 2$ and $t_1/t_2 = 1$ follows from Fig. 7
$c'_{12} = 1.4$
with $b_0/b_2 = 1.5$ *and* $t_0/t_2 = 0.5$ follows from Fig. 7
$c'_{02} = 0.1$

Choice of x :

For $x = 0$ we obtain $c_{12} = c'_{12}/0 = \infty$
$c_{02} = c'_{02}/(1-0) = c'_{02} = 0.1$
with $c_{12} = \infty$ and $s = 1$ follows from Figs. 4 or 5
$k'_1 = 5.13$
with $c_{02} = 0.1$ and $s = 1$ follows from Figs. 4 or 5
$k'_0 = 3.08$
$k'_1/k'_0 = 5.13/3.08 = 1.67$
for $x = 1$ we obtain $c_{12} = c'_{12}/1 = c'_{12} = 1.4$
$c_{02} = c'_{02}/(1-1) = \infty$
with $c_{12} = 1.4$ and $s = 1$ follows from Figs. 4 or 5
$k'_1 = 3.70$
with $c_{02} = \infty$ and $s = 1$ follows from Figs. 4 or 5
$k'_0 = 5.13$
$k'_1/k'_0 = 3.70/5.13 = 0.72$
The actual value for k'_1/k'_0 *is* 1.5 ; x therefore lies near zero.

1. *Approximation* $x = 0.1$; $1-x = 0.9$
$c_{12} = c'_{12}/x = 1.4/0.1 = 14$
$c_{02} = c'_{02}/(1-x) = 0.1/0.9 = 0.11$
with $c_{12} = 14$ and $s = 1$ follows from Figs. 4 or 5
$k'_1 = 4.47$
with $c_{02} = 0.12$ and $s = 1$ follows from Figs. 4 or 5
$k'_0 = 3.10$
$k'_1/k'_0 = 4.47/3.10 = 1.44$, *i.e.*, too small, x therefore is <0.1.

Fig. 14


SKETCH 1 OF § 212.123
Sections with webs which have different coefficients of restraint at the edges a & b.
k'- value of plate element 1
1/k'=(1/k'1 + 1/k'0)/2
read k'1 and k'0 from k'- diagrams with entries c0 or c0 and s=1.
read c0 and c0 from c- diagrams with entries b1/b2 and t1/t2 or b1/b2 and t1/t2.

2. *Approximation* $x = 0.05$; $1-x = 0.95$

$c_{12} = c'_{12}/x = 1.4/0.05 = 28$

$c_{02} = c'_{02}/(1-x) = 0.1/0.95 = 0.105$

$k'_1 = 4.63$

$k'_0 = 3.09$

$k'_1/k'_0 = 4.63/3.09 = 1.50$, *i.e.*, the desired value.

The fictitious slenderness ratio is therefore :

$$(l_k/i) = k'_1\,(b_1/t_1) = k'_0\,(b_0/t_0) = 4.63\times 10 = 3.09\times 15 = 46.3$$

2,1,2.1,2,3. *Sections with webs which have different coefficients of restraint at the edges a and b.* Examples for this type of unsymmetrical standard section are U, Z and H sections with unequal flanges or H sections where one of the half flanges is missing (Fig. 14).

The calculation of this type of unsymmetrical standard section is very simple. The coefficient of restraint c_a at the edge a and the coefficient of restraint c_b at the edge b are determined in accordance with the method already described. With these two coefficients of restraint the corresponding $k_a{}^1$ and $k_b{}^1$ values are taken from Figs. 2 or 3 (para. 1,2,1) and the average according to the following formula is taken :

$$1/k' = (1/k_{a1} + 1/k_b')/2$$

The fictitious slenderness ratio is then calculated as follows:—

$$(l_k/i) = k' \,.\, (b_1/t_1)$$

A calculated example for the unsymmetrical H section of Fig. 14 explains the procedure.

EXAMPLE 1 *OF PARA.* 2,1,2.1,2,3

Unsymmetrical H-section ; unequal coefficients of restraint of the web.

Restrained plate element 1 : (web)

$b_1 = 80$ cm. $t_1 = 2$ cm.

restraining plate element 2 : (flanges)

$b_2 = 10$ cm. $t_2 = 2$ cm.

restraining plate element 3 : (flanges)

$b_3 = 15$ cm. $t_3 = 3$ cm.

with $b_1/b_2 = 8$ and $t_1/t_2 = 1$ follows from Fig. 9

$c_a = 0.5\times 0.7 = 0.35$

with $c_a = 0.35$ and $s = 1$ follows from Fig. 1

$k'_a = 1.435$

with $b_1/b_3 = 5.33$ and $t_1/t_3 = 0.67$ follows from Fig. 9

$c_b = 0.5\times 0.11 = 0.055$

with $c_b = 0.055$ and $s = 1$ follows from Fig. 1

$k'_b = 1.30$

hence $1/k^1 = (1/1.435+1/1.30)/2 = 0.733$

$k' = 1/0.733 = 1.364$

the fictitious slenderness ratio is :

$(l_k/i) = 1.364\times 80/2 = 54.6$

This calculation shows that if an approximate k' is taken

$k' = (k_a' + k_b')/2$

practically the same result is obtained

$k' = (1.435 + 1.30)/2 = 1.3675$

This is due to the fact that in the given example k_a' and k_b' are very close. For the extreme case, however, where $c_a = \infty$ and, therefore, $k_a' = 1.63$, and where $c_b = 0$ and $k_b' = 1.24$ the average according to the above formula is 1.435, while in accordance with Table IV (para. 1,1,2) it should be 1.41 or more exactly 1.405.

2,1,2.1,2,4. *Exception to para.* 2,1,2.1,2,3 *applying to sections which have webs only restrained at one edge.* When analyzing sections in accordance with para. 2,1,2.1,2,3 it is possible to come across webs which are restrained at the edge b and are restraining at the edge a. The procedure given in the previous paragraph, however, can be applied in principle. The web is calculated first as if both edges had a coefficient of restraint c_b as for edge b. Then the corresponding k_b' value is calculated and the fictitious slenderness ratio is $(l_k/i)_b = k_b' \,.\, (b_1/t_1)$.

The web cannot be calculated a second time with a coefficient of restraint c_a at both edges as in para. 2,1,2.1,2,3 for the simple reason that this coefficient of restraint does not exist. The plate does not buckle, has no k_a' value and, therefore, no fictitious slenderness ratio $(l_k/i)_a$.

Instead of the web it is necessary, therefore, to consider the connected restrained and buckling plate in calculating the fictitious slenderness ratio $(l_k/i)_a'$.

The procedure in this case consists in calculating the average values on another basis than that of the k' values. Here we shall use average fictitious slenderness ratios.

$$(l_k/i) = [(l_k/i)_b + (l_k/i)_a']/2$$

This is necessary as on account of the different widths and thicknesses of the web and connected plate element the average values of k' do not make any sense. This procedure is strictly correct only if the critical buckling stresses are calculated with a straight-line formula. When the Euler formula is valid, or with buckling formulae of another type than the straight line, the average should be made on the basis of the critical buckling stresses.

The reason why this exception to para. 2,1,2.1,2,3 occurs lies in the simplifying assumption made in para. 1 that each individual plate which composes the section can be considered on its own and that the influence of adjacent plates can be expressed as coefficients of restraint.

As before the procedure will be explained with an example by analyzing the unsymmetrical H section as shown in Fig. 14.

EXAMPLE 1 *OF PARA.* 2,1,2.1,2,4

Unsymmetrical H-section ; exceptional case of para. 2,1,2.1,2,3

Edge b as in example 1 of para. 2,1,2.1,2,3

$k'_b = 1.30$

$(l_k/i)_b = 1.30\times 80/2 = 52.0$

Edge a :

restrained plate element 1 : (web)

$b_1 = 80$ cm. $t_1 = 2$ cm.

restraining plate element 2 : (flanges)

$b_2 = 20$ cm. $t_2 = 1$ cm.

with $b_1/b_2 = 4$ and $t_1/t_2 = 2$, no c-value can be obtained from Fig. 9 (para. 2,1,2.1,1,3). The half flanges are, therefore, not restraining but restrained at edge a.

We therefore put :

restrained plate element 1 : (flanges)

$b_1 = 20$ cm. $t_1 = 1$ cm.

restraining plate element 2 : (web)

$b_2 = 80$ cm. $t_2 = 2$ cm.

with $b_1/b_2 = 0.25$ and $t_1/t_2 = 0.5$ follows from Fig. 8 (para. 2,1,2.1,1,2)

$c_a = 2 \times 0.11 = 0.22$

with $c_a = 0.22$ and $s = 1$ follows from Figs. 4 or 5 (para. 1,2,1)

$k'_a = 3.19$

and finally the fictitious slenderness ratio is :

$$(l_k/i)'_a = k'_a \cdot b_1/t_1 = 3.19 \times 20/1 = 63.8$$

the determinant slenderness ratio for the web is therefore :

$$(l_k/i) = [(l_k/i)_b + (l_k/i)_a']/2 = (52.0 + 63.8)/2 = 57.9$$

Limiting case :

To analyze the conditions more fully at the edge a consider a limiting case in which the half flanges and the web have no influence on one another.

$c_a = \infty$

with $c_a = \infty$ and $s = 1$ follows from Fig. 1 (para. 1,2,1)

$k_a' = 1.63$

for this extreme case which never actually occurs, we obtain for the web :

$$1/k' = (1/k_a' + 1/k_b')/2 = (1/1.63 + 1/1.30)/2 = 0.689$$

$$k' = 1/0.689 = 1.452$$

finally the fictitious slenderness ratio for the web is :

$$(l_k/i) = k' \cdot b_2/t_2 = 1.452 \cdot 80/2 = 58.1$$

This value must be different from the value of 57.9 calculated above since the edge a is not freely supported.

Similar conditions arise with non-uniformly stressed plate elements when on one edge tensile stresses instead of compressive stresses occur. In this case as seen from Figs. 1 to 5 (para. 1,2,1) the k' values diminish and as in the example given above the fictitious slenderness ratio diminishes too.

2,1,2.1,2,5. *Rules for the use of the c diagrams*

When using the c digrams for unsymmetrical standard sections the c values must be given numerically. The procedure given in 2.1,2.1,1,5 which consists in indicating the interval only between two c curves is no longer permissible. The designer will have to get used to interpolating between c curves of the progression 1, 2, 5 or multiples of it. Should the inexperienced designer have some difficulty here, it is recommended that he draws between the curves given as many additional curves as he thinks necessary. Another but less convenient way out of the difficulty would be to make the interpolation graphically.

In the calculated examples given under paras. 2,1,2.1,2,1-1,2,2-1,2,3-1,2,4 only one c diagram has been used each time. This is a voluntary simplification of the examples but it is no condition for the validity of the method. In other words, the c values required in the calculations can be taken from any of the four c diagrams and they can be combined.

Sections with edges where more than three plates meet are calculated in a similar way. If, for instance, four plates meet the distinction must be made whether 1, 2 or 3 of the plates are restrained and correspondingly 3, 2 or 1 plates are restraining.

If the restrained plate is 1 and the 3 restraining plates 2, 3 and 4, then the following formula applies :—

$$1/c_1 = 1/c_{12} + 1/c_{13} + 1/c_{14}$$

If the two restrained plates are 1 and 0 and the two restraining plates are 2 and 3, then the following two formulae apply :—

$$1/c_1' = 1/c_{12} + 1/c_{13}$$

$$1/c_0' = 1/c_{02} + 1/c_{03}$$

Subsequently the x value or the $(1-x)$ value has to be determined by the trial and error method described in para. 2,1,2.1,2,2. The c_1' and c_0' must be divided by x or $(1-x)$ in order to obtain the real coefficients of restraint of the plates 1 and 0.

In the case of three restrained plates 1, 0 and 00 and one restraining plate 2, the following formulae are valid :—

$$c_{12}' = c_{12} \quad c_{02}' = c_{02} \quad c_{002}' = c_{002}$$

Subsequently, the restraint which is produced by plate 2 must be distributed among the plates 1, 0 and 00 with a similar method of trial and error as already described. An example will show that the distribution of the restraint among three plates is less difficult than it appears.

EXAMPLE 1 OF PARA. 2,1,2.1,2,5

Section with edges where 4 different plate elements meet and where three plate elements are restrained

Restrained plate element 1 : (flange)

$b_1 = 10$ cm. $t_1 = 1$ cm.

co-restrained plate element 0 : (flange)

$b_0 = 12$ cm. $t_0 = 1$ cm.

co-restrained plate element 00 : (web)

$b_{00} = 60$ cm. $t_{00} = 2$ cm.

restraining plate element 2 : (web)

$b_2 = 20$ cm. $t_2 = 1$ cm.

with $b_1/b_2 = 10/20 = 0.5$ and $t_1/t_2 = 1$ follows from Fig. 8 (para. 2,1,2.1,1,2)

$c'_{12} = 1.5$

with $b_0/b_2 = 12/20 = 0.6$ and $t_0/t_2 = 1$ follows from Fig. 8 (para. 2,1,2.1,1,2)

$c'_{02} = 1.2$

with $b_{00}/b_2 = 60/20 = 3$ and $t_{00}/t_2 = 2$ follows from Fig. 10 (para. 2,1,2.1,1,4)

$c'_{002} = 4.2$

according to para. 2,1,2.1,2,2, the following conditions must be satisfied :—

$k_1'/k'_0 = (b_0/t_0)/(b_1/t_1) = 12/10 = 1.2$
$k'_1/k'_{00} = (b_{00}/t_{00})/b_1/t_1) = 30/10 = 3$
$k'_0/k'_{00} = (b_{00}/t_{00})/b_0/t_0) = 30/12 = 2.5$ [= 3/1.2]

Choice of x, y & z

The multipliers $1/x$, $1/y$ and $1/z$ of c'_{12}, c'_{02} and c'_{002} must be chosen in such a way that the above equations are satisfied and that

$x + y + z = 1$

From Fig. 1 (para. 1,2,1) with a variation of z between 1 and 0 the k'_{00}-value varies only between 1.60 and 1.63. Therefore, z can be chosen almost at random.

Therefore :

z	$= 0$	*or*	$z = 1$
c_{00}	$= 4.2/0 = \infty$		$c_{00} = 4.2$
k'_{00}	$= 1.63$		$k'_{00} = 1.60$

Hence,

$k'_1 = 3 \times 1.63 = 4.89$ *or* $k'_1 = 3 \times 1.6 = 4.8$
$k'_0 = 2.5 \times 1.63 = 4.07$ $k'_0 = 2.5 \times 1.6 = 4.0$

From Figs. 4 or 5 (para. 1,2,1) follow the corresponding c-values

$c_1 = 150$ *or* $c_1 = 80$
$c_0 = 3.6$ $c_0 = 3.2$

From which it can be seen that the c_1-value can also have a considerable variation without affecting the result. We begin, therefore, by determining the values x, y and z with $c_0 = 3.4$ which is the average of the above-mentioned values.

$c_0 = c'_{02}/y$ *or* $y = 1.2/3.4 = 0.353$

with $c_1 = (150+80)/2 = 115$ then
$c_1 = c'_{12}/x$ *or* $x = 1.5/115 = 0.013$

and finally :

$c_{00} = c'_{002}/z$ where
$z = 1 - x - y = 0.634$

The distribution of the restraints produced by plate 2 can therefore be approximately determined from the above data.

Check

$c_0 = 1.2/0.353 = 3.4$; $k'_0 = 4.03$
$c_1 = 1.5/0.013 = 115$; $k'_1 = 4.85$
and $k'_1/k'_0 = 4.85/4.03 = 1.203$ [should be 1.2]
$c_{00} = 4.2/0.634 = 6.6$; $k'_{00} = 1.62$
and $k'_1/k'_{00} = 4.85/1.62 = 2.995$ [should be 3]
$k'_0/k'_{00} = 4.03/1.62 = 2.488$ [should be 2.5]

In para. 2,1,2.1,2 all the cases which may arise have not been represented systematically, but the designer will find for himself the way in which a given section has to be calculated.

Finally, it should be pointed out that in the k' and c diagrams only quantities without dimensions appear. This is due to the desire to make these diagrams usable for the metric as well as for the Imperial system, since the author's organization works in both of them. For the same reason fictitious slenderness ratios are used instead of direct calculation of the critical buckling stresses which must be expressed in kg. per sq. cm. or tons per sq. in. or lb. per sq. in. The dimensions in cm. given in the calculation examples can, therefore, also be in inches without changing the results.

2,1,2.1,3. *Special Sections.* What has been calculated so far can now be summarized. The c values of the four basic types of standard sections are known and the shapes which can be derived from these four basic types and for which c values are either halved or doubled have been indicated. Different types of unsymmetrical standard sections have also been analyzed where the determinant c, k' or (l_k/i) values are obtained with an intermediary calculation. All the sections investigated have one feature in common ; the plate elements meeting at any edge are at right-angles or they lie in identical planes to both sides of the edge.

The indicated calculation method can now be applied to sections with edges where the plate elements are no longer at right-angles ; such sections are designated as " special sections." These special sections are calculated in exactly the same way as outlined in paras. 2,2,2.1,1 and 2,1,2.1,2 provided that the edges are well defined. These edges have, as assumed in the theory of plate buckling, the function of preventing the formation of buckles. They, therefore, must remain straight. This is not possible with sections which can be derived from angle sections where the plate elements form only a very small angle. Such open angles where the two legs are almost in line have no well-defined edge and, therefore, cannot be considered as a combination of two plate elements since the edge is missing. Such sections fail through general buckling.

Also small lips at edges are, under certain conditions, not sufficient to keep the edges straight. In this case it is not permitted, when studying local buckling, to take account of them. Edges with such small lips are to be considered as free edges, *i.e.*, edges which are not supported.

As can be inferred from the above, for the calculation of special sections a trial and error method is used by which is determined which of the plate elements meeting at an edge are restrained and which ones are restraining. The experienced designer will have no difficulty at all in getting

quick results. It would mean considerable loss of time—assuming that the designer is capable of carrying out the calculations—to analyze these special sections according to the method referred to in para. 1. In this method, as already said, all plate elements which constitute a section are considered as influencing all others and in this method naturally the solution is obtained without trial and error.

The calculation of special sections for local buckling is particularly important with aluminium as the designer is very frequently called upon to develop special sections meeting his particular requirements. Also workshops very often produce their own sections by welding up standard sections or plate, or by brake pressing plate. Such sections should not fail prematurely on account of local buckling.

Naturally the local buckling is equally important with standard extruded sections but here the producer of semi-finished products has it in his power to choose the wall thickness of his standard sections in such a way that local buckling does not occur with the usual slenderness ratios for general buckling.

Proposals for government specifications regarding dimensions of standard sections have been made where the wall thicknesses of these sections have been deliberately chosen thick enough that the designer does not have to worry about local buckling of the plate elements. The author feels that such a procedure of solving the problem of local buckling is not adequate as it entails waste of material and jeopardizes the economics of aluminium against other materials.

2,1,2.2 *Non-uniform Stress Distribution*

After having calculated or at least indicated all possible cases which might arise with uniform stress distribution, the method must now extend to non-uniform stress distribution. The influence of non-uniform stress distribution can be taken into account in the formulae of para. 2,1,2.1,1,1-1,1,2-1,1,3-1,1,4 if the coefficient g as given in Table I of para. 2,1,2 is altered appropriately.

This coefficient g can be taken from nomogram 1 of para. 2,1,2.2 for the various types of plates, the various combinations of plates and for variable stress ratios s_1 and s_2 within the limits of $+1$ and -1. When using this nomogram, the application of the formula given in para. 2,1,2 becomes unnecessary. For the above given limits of stress ratio—within which local buckling may usually be expected—the coefficients g vary between the values indicated in Table VIII of para. 2,1,2.2. For these limits, nomogram 1 of para. 2,1,2.2 has been established.

Taking into account the fact that in the formula for the calculation of the c values the magnitude of d_F or d_S—which are used in the calculation of the half wave-lengths of the buckles—varies only very slightly with the stress ratio, it is permitted, in order to simplify the calculations, to consider d_F or d_S as constants corresponding to the values used with uniform stress distribution. Hence, an assumption is made similar to that which Bleich has made for the calculation of the c values for stress ratio $s = 1$.

Bleich puts for flanges $d_F = \infty$ = constant, for the range from 1.668 to ∞ and for webs $d_S = 1$ = constant, for the range from 1 to 0.668. With

Fig. 15

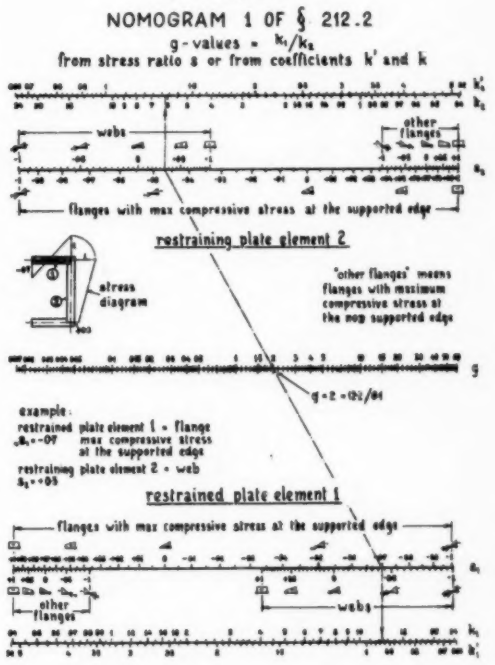

NOMOGRAM 1 OF § 212.2
g - values = k1/k2
from stress ratio s or from coefficients k' and k
webs
other flanges
flanges with max compressive stress at the supported edge
restraining plate element 2
stress diagram
'other flanges' means flanges with maximum compressive stress at the non supported edge
example:
restrained plate element 1 = flange
max compressive stress at the supported edge
restraining plate element 2 = web
restrained plate element 1
flanges with max compressive stress at the supported edge
other flanges
webs

TABLE VIII (para. 2,1,2.2)—*Limiting g-values with Non-uniform Stress Distribution*

Type of plate element and combination of elements	Minimum g	Maximum g
flange/flange	0.406/23.67 = 0.01715	23.67/0.406 = 58.300
flange/web	0.406/23.88 = 0.01700	23.67/ 4 = 5.918
web/flange	4 /23.67 = 0.1690	23.88/0.406 = 58.818
web/web	4 /23.88 = 0.1675	23.88/ 4 = 5.970

TABLE IX (para. 2,1,2.2)—*Coefficients d_F or d_s with Non-uniform Stress Distribution*

Type of plate elements	Coefficient of restraint c	Coefficients d_F and d_S for the stress ratios : 1	0.5	0	−0.5	−1	d_p or d_s used by author
Web	∞	1	0.998	0.983	0.915	0.667	1
identical c	5	0.968					0.968
at both edges	1	0.878					0.878
	0.2	0.755					0.755
	0.05	0.701					0.701
	0	0.668	0.660	0.653	0.608	0.473	0.668
Flange	∞	∞	∞	∞	∞	∞	∞
max. compressive	200	7.59					7.59
stress	50	5.53					5.53
at the supported	10	3.62					3.62
edge	2	2.60					2.60
	0.5	1.98					1.98
	0.1	1.72					1.72
	0	1.668		1.58		1	1.668
Flange	∞	∞	∞	∞	∞	∞	∞
max. compressive	200	7.59					7.59
stress	50	5.53					5.53
at the non-supported	10	3.62					3.62
edge	2	2.60					2.60
	0.5	1.98					1.98
	0.1	1.72					1.72
	0	1.668		1.667		1.63	1.668

the first assumption, the errors in the k' or k values are noticeable as can be seen from Fig. 11 (para. 2,1,2.1,1,6). With the second assumption, however, no errors can be detected with respect to the exact calculation.

The extreme values for the simplifications can be taken from Table IX (para. 2,1,2.2). The errors are not bigger than those incurred by Bleich with his second assumption. This table is not complete as the information available in the technical literature is fragmentary. However, the figures known to us give a sufficiently accurate picture of the extent of our simplifications.

The above principles have been applied to the calculation of the c diagrams in the sub-para. 2,1,2.2,1-2,2-2,3-2,4. For each of the four basic types of standard sections nine diagrams for nine different g values have been calculated. The limits of g are given in Table I of para. 2,1,2.2. Between these two limits lies the already known diagram for uniform stress distribution $s_1 = 1$ and $s_2 = 1$. Between these three diagrams six additional diagrams have been placed in such a way that the straight line for $c = \infty$ covers the whole field as regularly as possible. For this reason the chosen g values are not round figures.

2,1,2.2,1. *ANGLE SECTIONS.* The g values used in the diagrams for angle sections are given

TABLE X (para. 2,1,2.2,1)—*g-values with Variable s for the plate-combination flange/flange*

g min.	= 0.0172 . $b_1/b_2 = 0.131$.	t_1/t_2
	(Fig. 16, para. 2,1,2.2,1)	
	0.121 . $b_1/b_2 = 0.348$.	t_1/t_2
	(Fig. 17, para. 2,1,2.2,1)	
	0.320 . $b_1/b_2 = 0.5658$.	t_1/t_2
	(Fig. 18, para. 2,1,2.2,1)	
	0.613 . $b_1/b_2 = 0.783$.	t_1/t_2
	(Fig. 19, para. 2,1,2.2,1)	
	1* . $b_1/b_2 =$	t_1/t_2
	(Fig. 20, para. 2,1,2.2,1)	
	1.632 . $b_1/b_2 = 1.278$.	t_1/t_2
	(Fig. 21, para. 2,1,2.2,1)	
	3.126 . $b_1/b_2 = 1.768$.	t_1/t_2
	(Fig. 22, para. 2,1,2.2,1)	
	8.238 . $b_1/b_2 = 2.870$.	t_1/t_2
	(Fig. 23, para. 2,1,2.2,1)	
g max.	=58.3 . $b_1/b_2 = 7.635$.	t_1/t_2
	(Fig. 24, para. 2,1,2.2,1)	

* Value for $s_1 = 1$ and $s_2 = 1$

TABLE XI (para. 2,1,2.2,2)—*g-values with Variable s for the plate-combination flange/web*

g min.	= 0.0170 . $b_1/b_2 = 0.1304$.	t_1/t_2
	(Fig. 25, para. 2,1,2.2,2)	
	0.0373 . $b_1/b_2 = 0.1931$.	t_1/t_2
	(Fig. 26, para. 2,1,2.2,2)	
	0.0655 . $b_1/b_2 = 0.2559$.	t_1/t_2
	(Fig. 27, para. 2,1,2.1,2)	
	0.1015* $b_1/b_2 = 0.3186$.	t_1/t_2
	(Fig. 28, para. 2,1,2.2,2)	
	0.1487 . $b_1/b_2 = 0.3856$.	t_1/t_2
	(Fig. 29, para. 2,1,2.2,2)	
	0.2385 . $b_1/b_2 = 0.4884$.	t_1/t_2
	(Fig. 30, para. 2,1,2.2,2)	
	0.4432 . $b_1/b_2 = 0.6657$.	t_1/t_2
	(Fig. 31, para. 2,1,2.2,2)	
	1.093 . $b_1/b_2 = 1.0454$.	t_1/t_2
	(Fig. 32, para. 2,1,2.2,2)	
g max.	= 5.918 . $b_1/b_2 = 2,433$.	t_1/t_2
	(Fig. 33, para. 2,1,2.2,2)	

* Value for $s_1 = 1$ and $s_2 = 1$

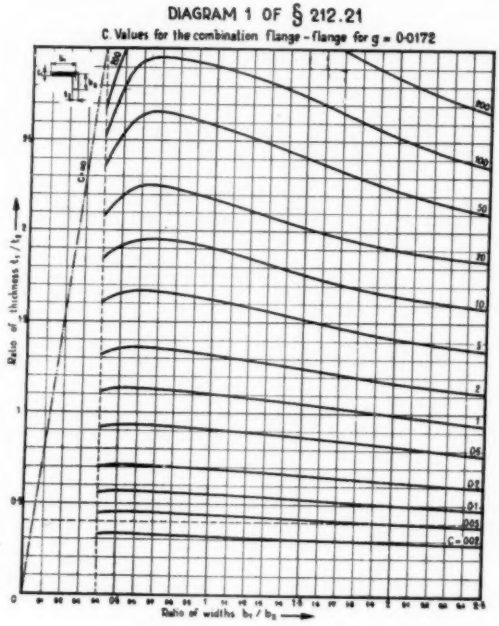


DIAGRAM 1 OF § 212.21
C. Values for the combination flange - flange for g = 0·0172
Ratio of thickness t_1 / t_2
Ratio of widths b_1 / b_2
C=0·02
C=0·01
C=0·005
C=0·002

Fig. 16

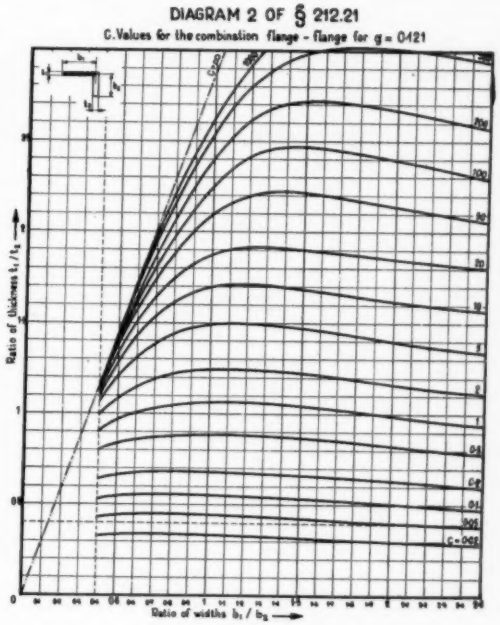


DIAGRAM 2 OF § 212.21
C. Values for the combination flange - flange for g = 0·121
Ratio of thickness t_1 / t_2
Ratio of widths b_1 / b_2
C=0·02
C=0·01
C=0·005
C=0·002

Fig. 17

Fig. 18

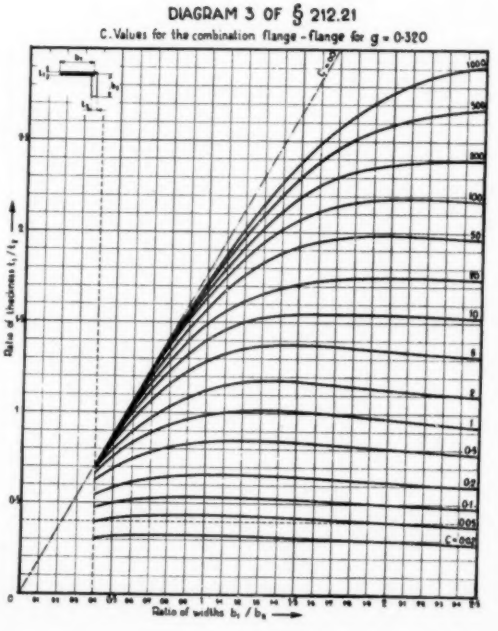


DIAGRAM 3 OF § 212.21
C. Values for the combination flange - flange for g = 0·320
Ratio of thickness t_1 / t_2
Ratio of widths b_1 / b_2
C=0·02
C=0·01
C=0·005
C=0·002

Fig. 19

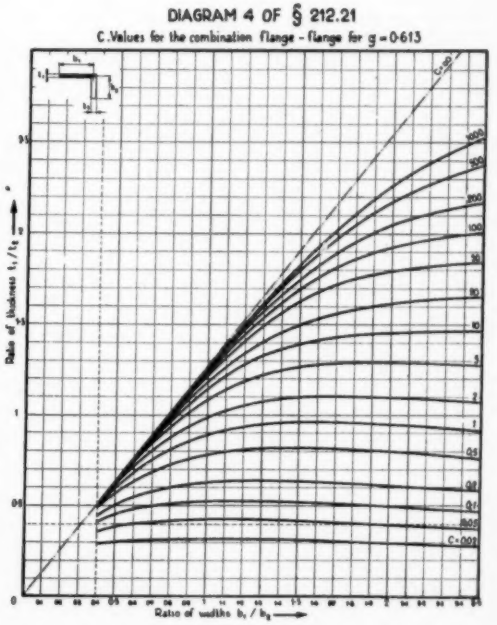


DIAGRAM 4 OF § 212.21
C. Values for the combination flange - flange for g = 0·613
Ratio of thickness t_1 / t_2
Ratio of widths b_1 / b_2
C=0·02
C=0·01
C=0·005
C=0·002

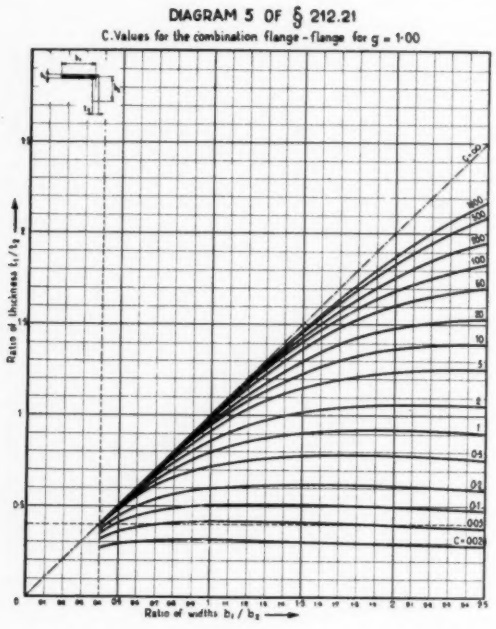


Fig. 20

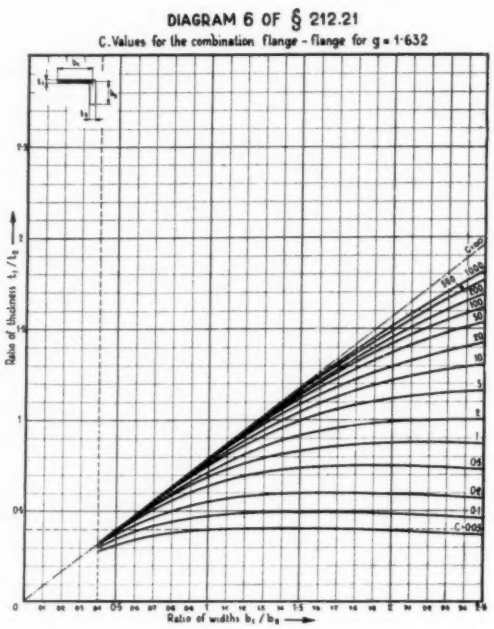
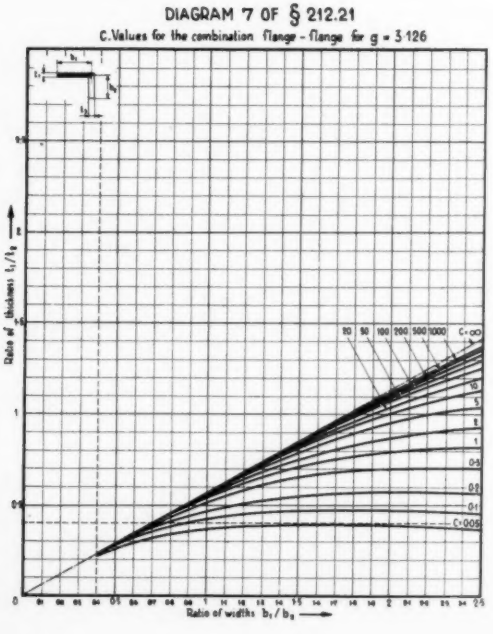


Fig. 21

Fig. 22

Fig. 23



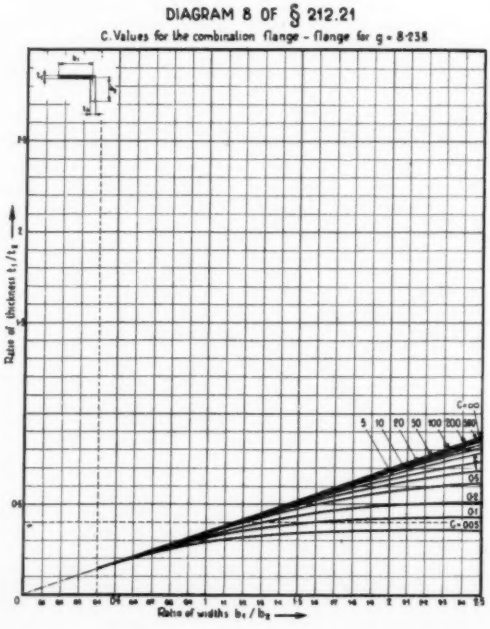


TABLE XII (para. 2,1,2.2,3)—*g-values with Variable s for the plate-combination web/flange*

g min.	= 0.1690 . b_1/b_2 = 0.4111 .	t_1/t_2
	(Fig. 34, para. 2,1,2.2,3)	
	1.1946 . b_1/b_2 = 1.093 .	t_1/t_2
	(Fig. 35, para. 2,1,2.2,3)	
	3.1506 . b_1/b_2 = 1.775 .	t_1/t_2
	(Fig. 36, para. 2,1,2.2,3)	
	6.037 . b_1/b_2 = 2.457 .	t_1/t_2
	(Fig. 37, para. 2.1,2.2,3)	
	9.852* . b_1/b_2 = 3.139 .	t_1/t_2
	(Fig. 38, para. 2,1,2.2,3)	
	13.562 . b_1/b_2 = 3.683 .	t_1/t_2
	(Fig. 39, para. 2,1,2.2,3)	
	19.841 . b_1/b_2 = 4.454 .	t_1/t_2
	(Fig. 40, para. 2,1,2.2,3)	
	31.758 . b_1/b_2 = 5.635 .	t_1/t_2
	(Fig. 41, para. 2,1,2.2,3)	
g max.	=58.818 . b_1/b_2 = 7.669 .	t_1/t_2
	(Fig. 42, para. 2,1,2.2,3)	

*Value for $s_1 = 1$ and $s_2 = 1$

TABLE XIII (para. 2,1,2.2,4)—*g-values with Variable s for the plate-combination web/web*

g min.	= 0.1675 . b_1/b_2 = 0.4093 .	t_1/t_2
	(Fig. 43, para. 2,1,2.2,4)	
	0.3675 . b_1/b_2 = 0.6062 .	t_1/t_2
	(Fig. 44, para. 2,1,2.2,4)	
	0.6450 . b_1/b_2 = 0.8031 .	t_1/t_2
	(Fig. 45, para. 2,1,2.2,4)	
	1* . b_1/b_2 =	t_1/t_2
	(Fig. 46, para. 2,1,2.2,4)	
	1.286 . b_1/b_2 = 1.134 .	t_1/t_2
	(Fig. 47, para. 2,1,2.2,4)	
	1.715 . b_1/b_2 = 1.309 .	$t_1/$
	(Fig. 48, para. 2,1,2.2,4)	
	2.399 . b_1/b_2 = 1.549 .	
	(Fig. 49, para. 2,1,2.2,4)	
	3.595 . b_1/b_2 = 1.896 .	t_1/t_2
	(Fig. 50, para. 2,1,2.2,4)	
g max.	= 5.970 . b_1/b_2 = 2.443 .	t_1/t_2
	(Fig. 51, para. 2,1,2.2,4)	

*Value for $s_1 = 1$ and $s_2 = 1$

in Table X. In this table the equation for the straight line valid for $c = \infty$ is also given.

2,1,2.2,2. *FLANGES OF U SECTIONS.* The g values used for the calculation of the diagrams for the flanges of U sections are given in Table XI. This table also gives the equation for the straight line valid for $c = \infty$.

2,1,2.2,3. *WEBS OF U SECTIONS.* The g values used for calculating the webs of U sections are given in Table XII. As before, this table also gives the equation of the straight line $c = \infty$.

2,1,2.2,4. *RECTANGULAR TUBES.* The g values used for the calculation of the rectangular tubes are given in Table XIII, which also gives the equations of the straight line for $c = \infty$.

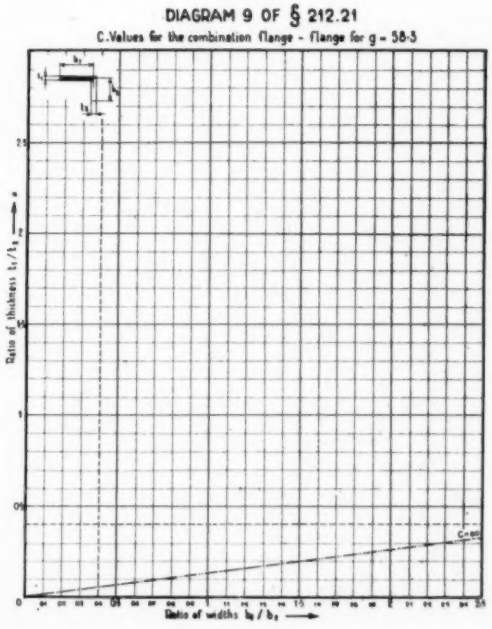
2,1,2.2,5. *RULES FOR THE USE OF THE c DIAGRAMS.* The 36 diagrams given for the c values when the stress distribution is not uniform are used in the same way as the four diagrams for the c values for uniform stress distribution. When extending their use to symmetrical standard sections which are derived from the basic types, attention must be paid to the fact that the symmetry not only applies to the shape of the section but also to the stress distribution. Geometrically symmetrical standard sections must, therefore, be considered in certain circumstances as unsymmetrical standard sections.

Between the nine diagrams of a given basic type of section, the coefficient g which corresponds to the actual stress distribution must be interpolated. This interpolation is not always easy to carry out and would be admittedly more correct if more than nine diagrams were given. However, the desire to keep the method simple necessitates no increase in the number of these diagrams. The author believes that the arrangement chosen represents an acceptable compromise for the designer.

This interpolation is carried out in the following way. First, determine the c values in the two diagrams for the two g values which are situated immediately above or below the actual g value. Subsequently, c is interpolated linearly. For the exceptional case where one of the c values becomes ∞ or that it is positioned in the neighbourhood of the straight line $c = \infty$, a third g value must be used in order to give the variability of c with g more accurately than with a linear law.

(*Text continued in page* 50)

Fig. 24



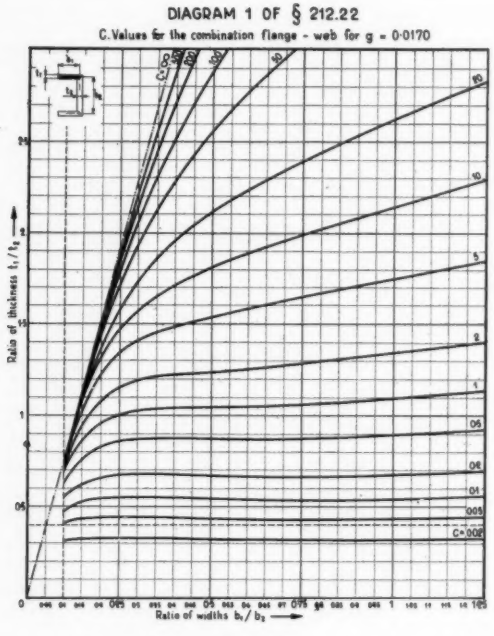


Fig. 25

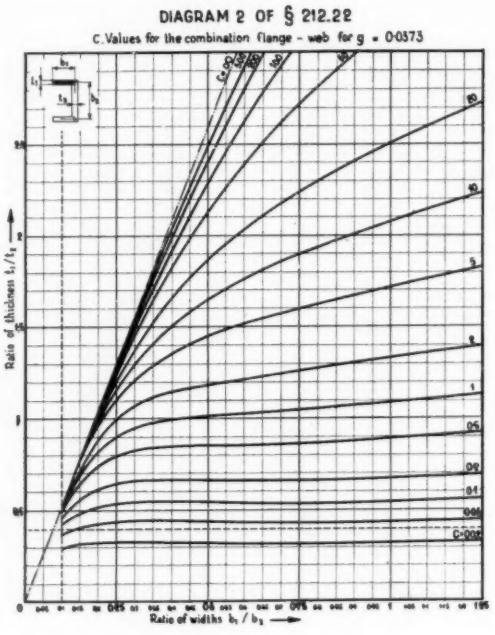
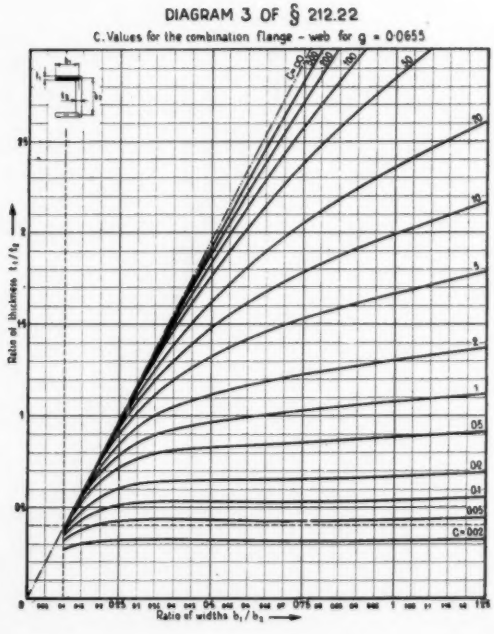
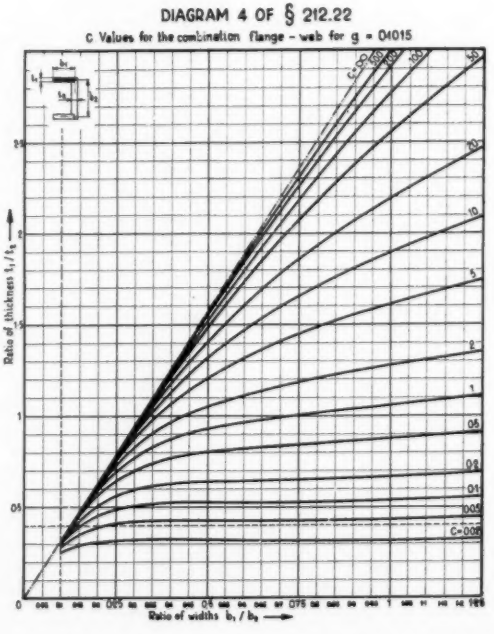


Fig. 26

Fig. 27

Fig. 28





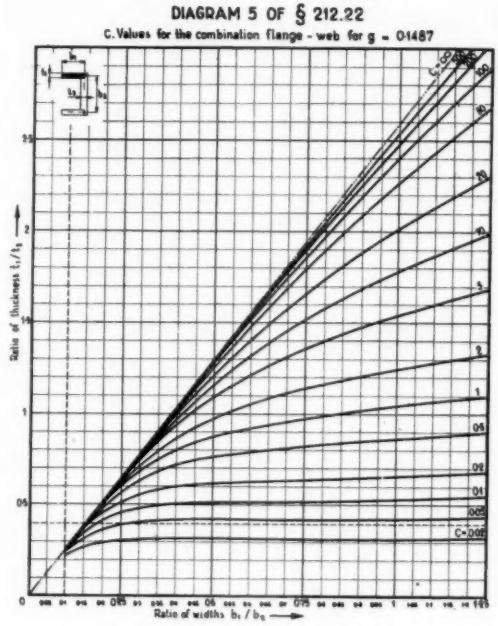


Fig. 29

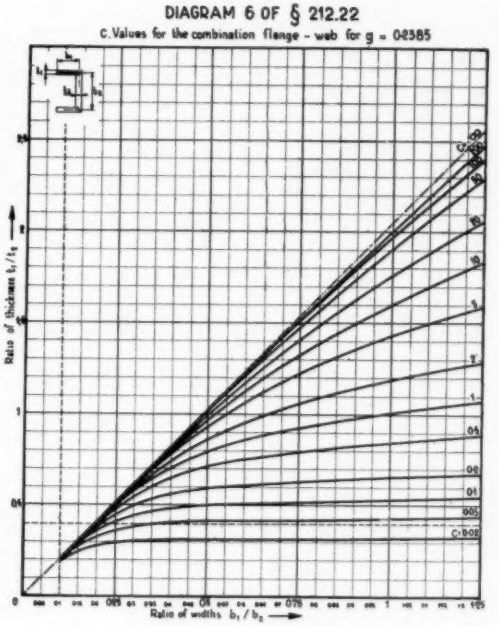


Fig. 30

Fig. 31

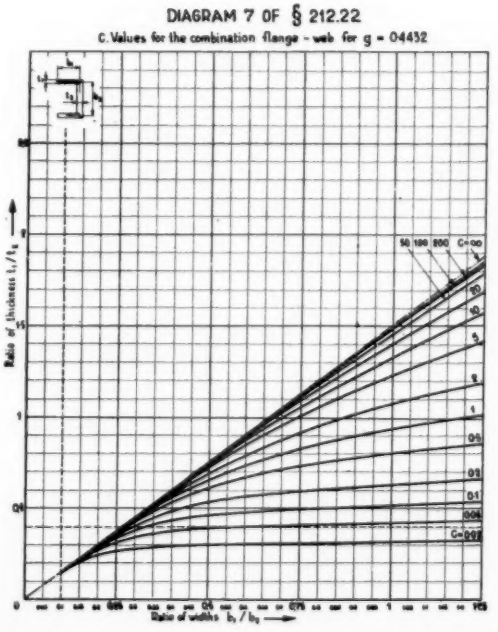
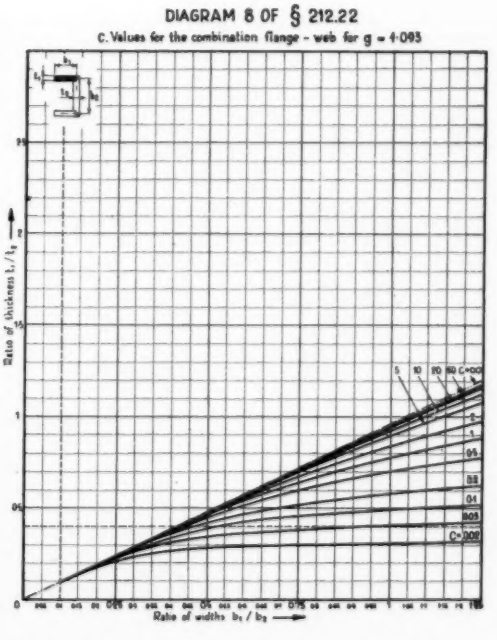


Fig. 32



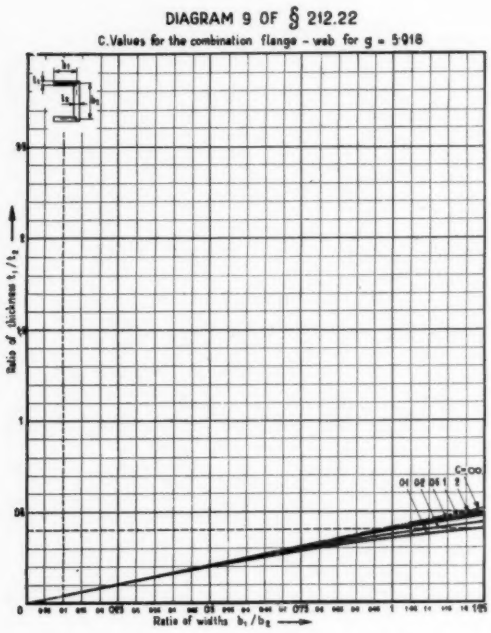


DIAGRAM 9 OF § 212.22
C. Values for the combination flange - web for g = 5·918
Ratio of thickness t_1/t_2
Ratio of widths b_1/b_2
C=∞
0·1 0·2 0·5 1 2 5

Fig. 33

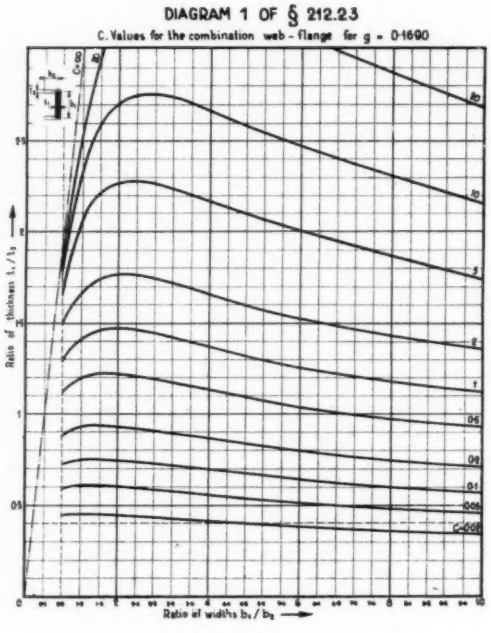


Fig. 34

Fig. 35

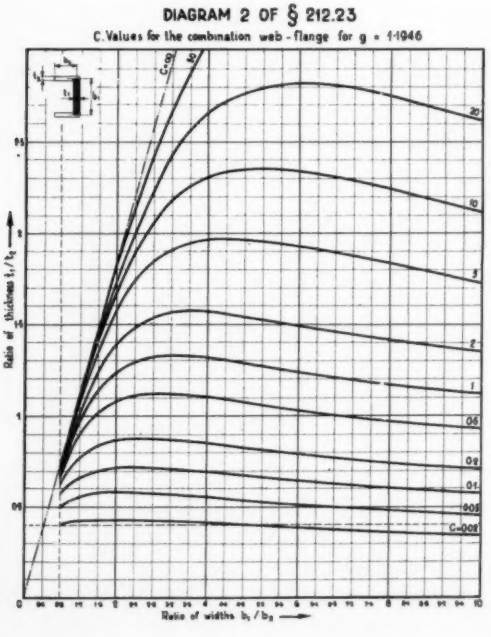
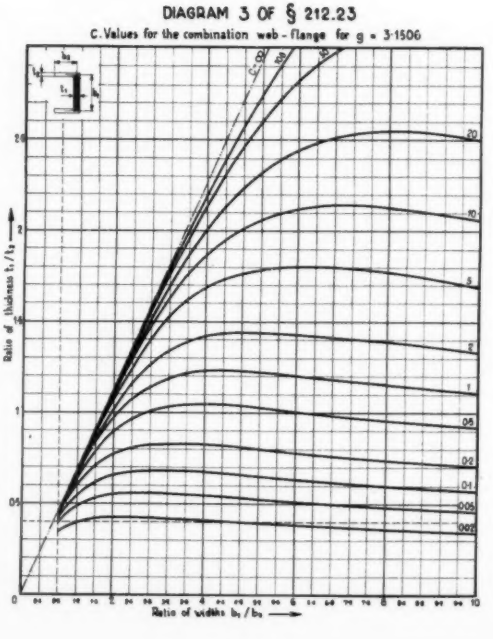


Fig. 36



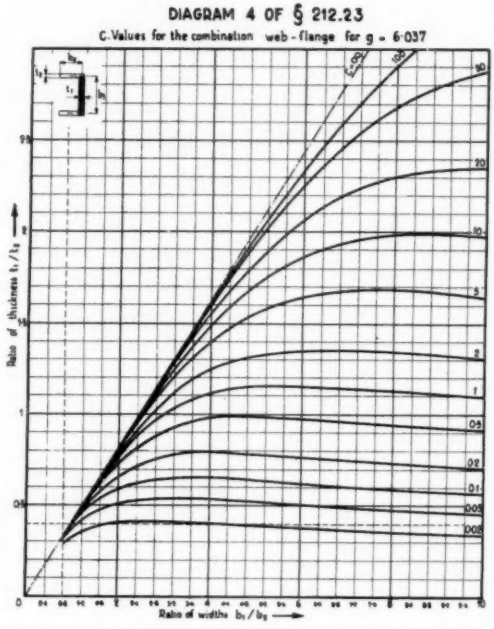


Fig. 37

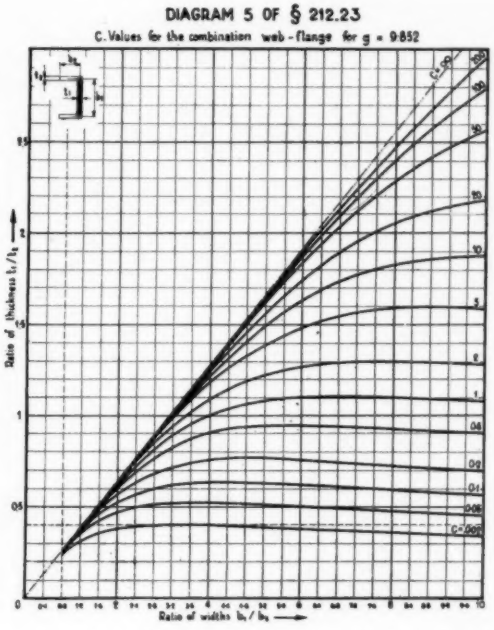


Fig. 38

Fig. 39

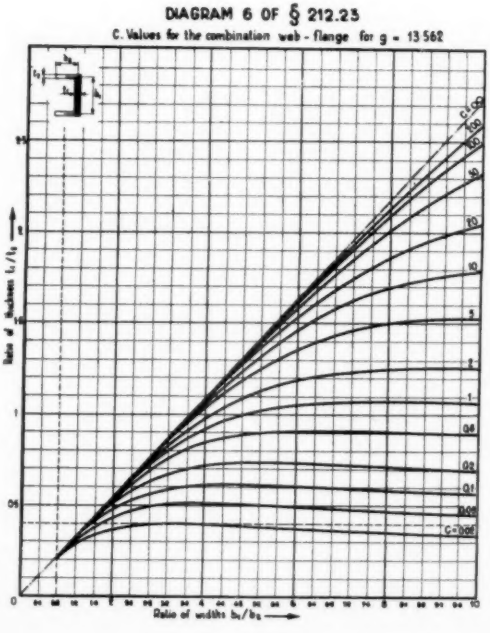
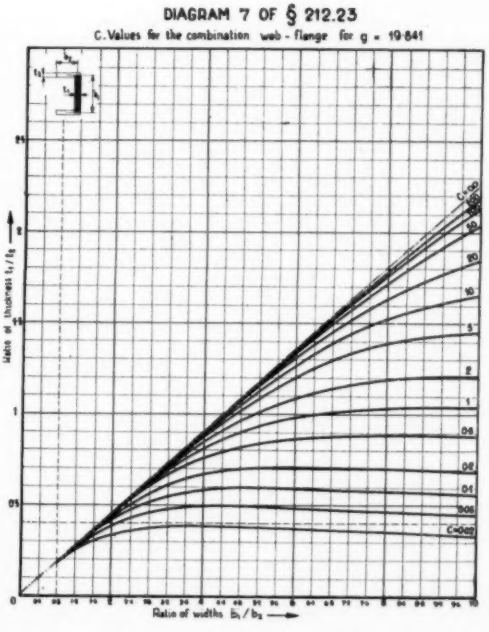


Fig. 40



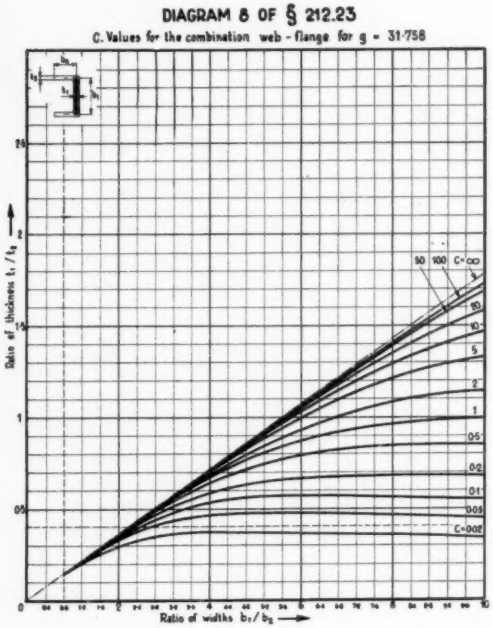


Fig. 41

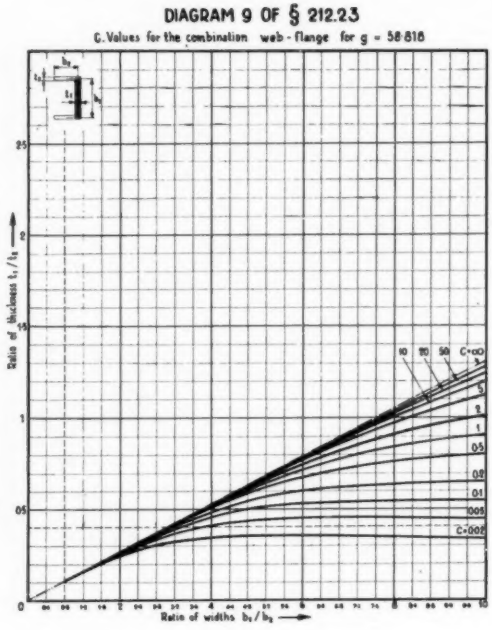


Fig. 42

Fig. 43

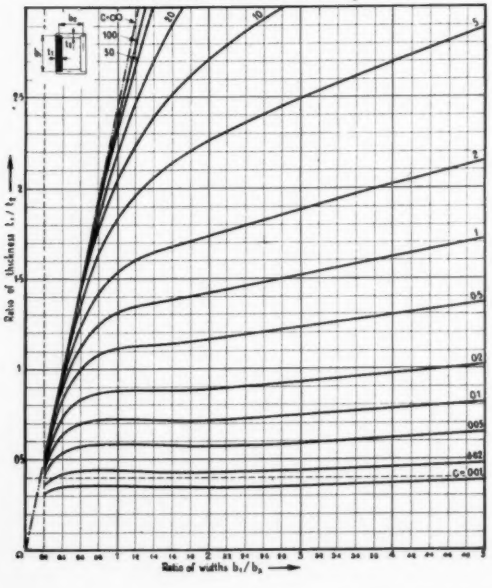
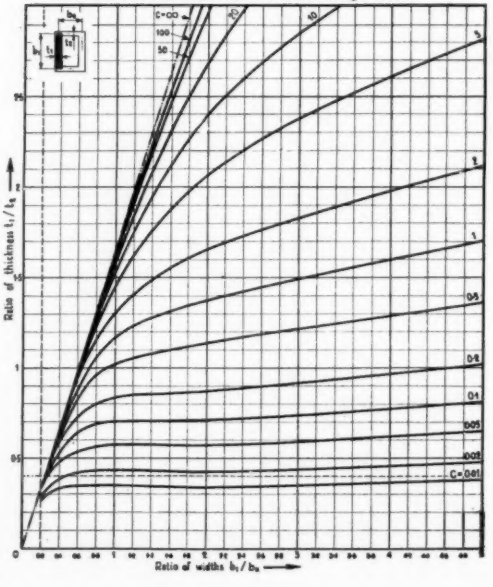


Fig. 44



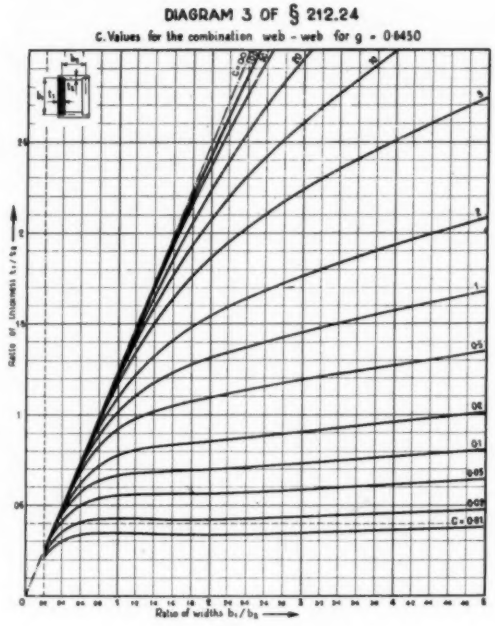


Fig. 45

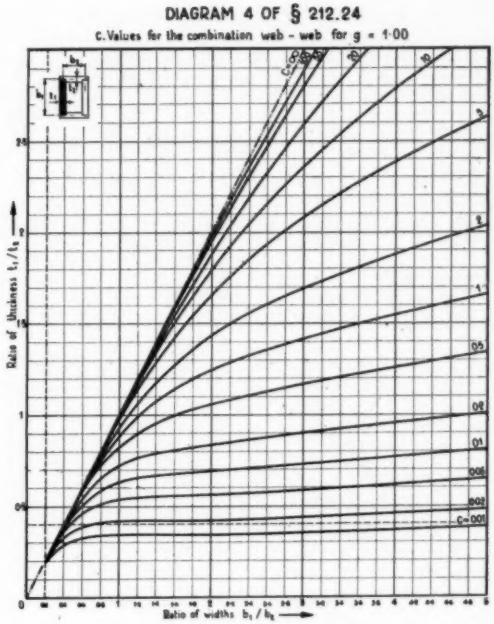


Fig. 46

Fig. 47

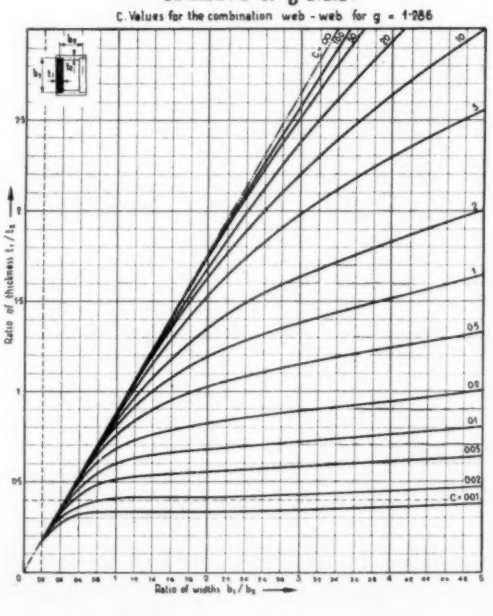
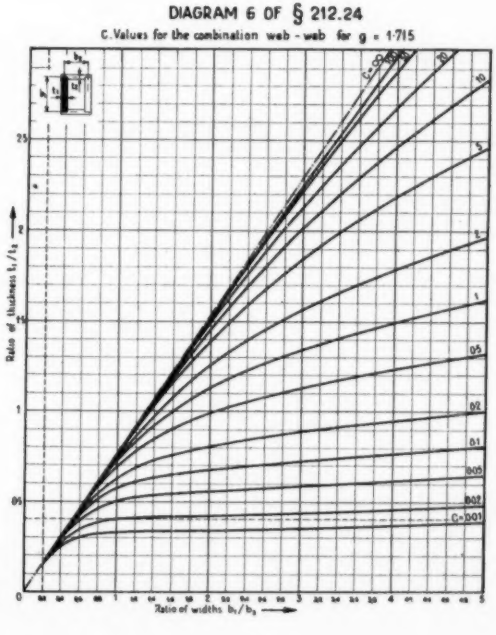


Fig. 48



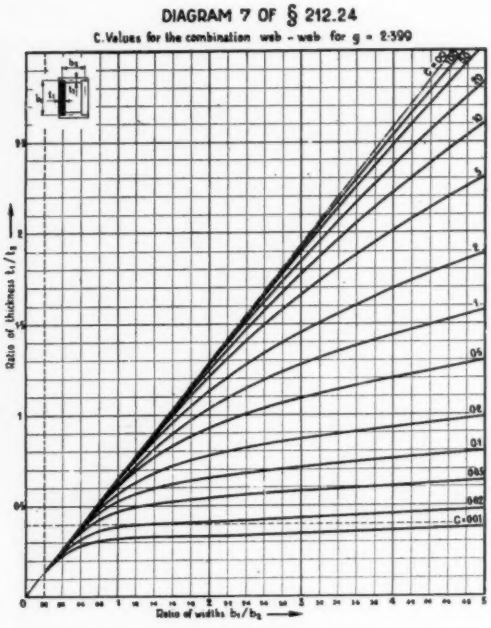


Fig. 49

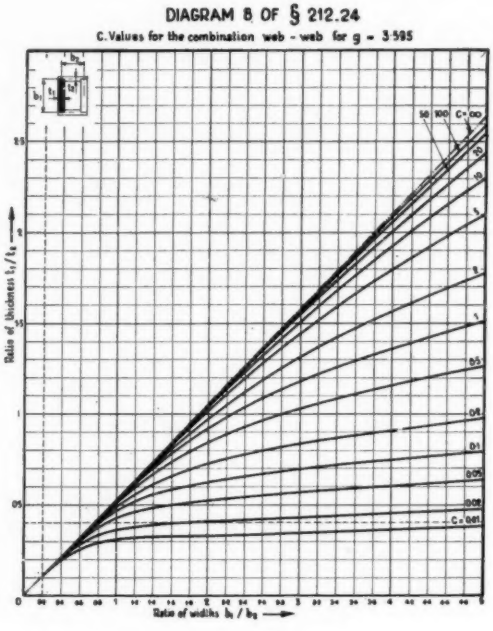
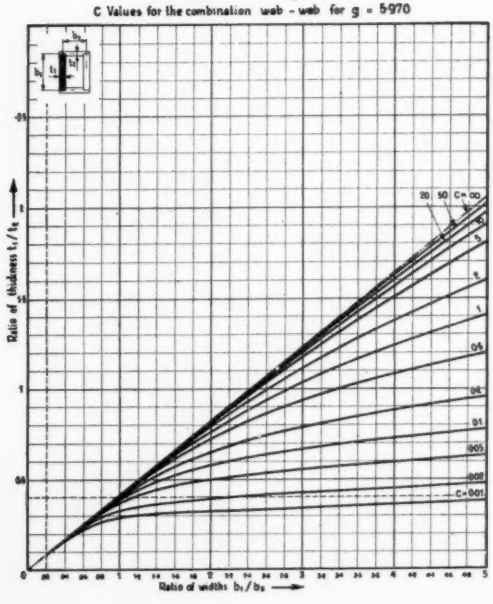


Fig. 50

Fig. 51



The example below illustrates the procedure. The section studied is the U-section shown in the nomogram of para. 2,1,2.2 (Fig. 15).

EXAMPLE 1 *OF PARA.* 2,1,2.2,5

Interpolation between Figs. 16 *to* 51 (*paras.* 2,1,2.2,1-2,2-2,3-2,4.

Restrained plate element 1 : (flange)

$b_1 = 40$ cm. $t_1 = 1$ cm. $s_1 = -0.7$

restraining plate element 2 : (web)

$b_2 = 40$ cm. $t_2 = 2$ cm. $s_2 = +0.3$

from s_1 and s_2 we get for the combination of plate elements flange/web and the given position of the greatest compressive stress from the nomogram (Fig. 15, para. 2,1,2.2)

$g = 2.0$

We must therefore determine (*see* Table XI (para. 2,1,2.2,2)] the c-values for the following two g-values : $g_{II} = 1.093$ and $g_{IV} = 5.918$. In order to improve the interpolation we have to take yet a third value : $g_I = 0.4432$.

From the Figs. 30, 31 and 32 (para. 2,1,2.2,2) it follows for the entries $b_1/b_2 = 1$ and $t_1/t_2 = 0.5$:

$c_I = 0.09$

$c_{II} = 0.105$

$c_{III} = \infty$

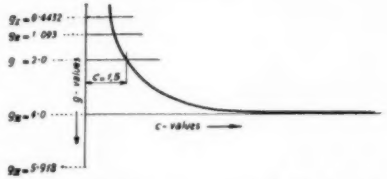
c_{IV} = does not exist, plate element 2 restrained.

Between the values c_{II} and c_{IV} we cannot interpolate since c_{IV} does not exist. We calculate,

therefore a g-value for which $c_{III} = \infty$. This occurs if according to para. 2,1,2

$$1 - g \cdot (t_1/t_2)^2/(b_1/b_2)^2 = 0 \quad or$$
$$g_{III} = (b_1/b_2)^2/(t_1/t_2)^2 = 1^2/0.5^2 = 4$$

The interpolation, which in this case is best made graphically, gives : $c = 1.5$



From Fig. 4 (para. 1,2,1) it follows with the entries $s = -0.7$ and $c = 1.5$ and since the greatest compressive stress occurs at the supported edge :

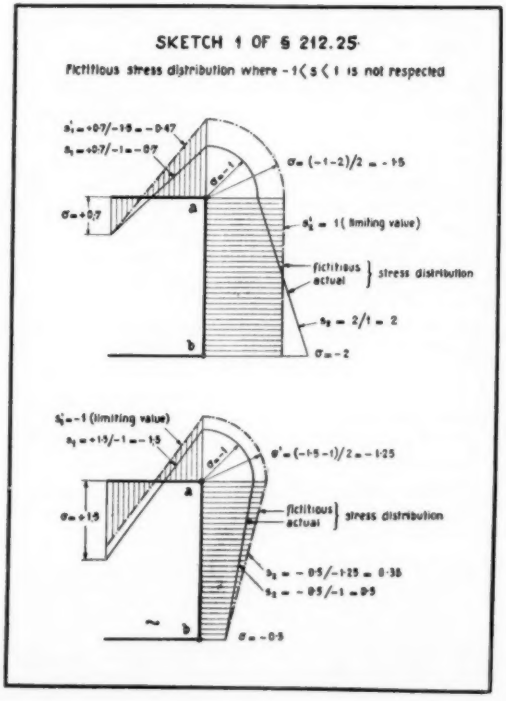
$k' = 0.72$

the corresponding fictitious slenderness ratio is :

$(l_k/i) = 0.72 \times 40/1 = 28.8.$

Finally, it is necessary to point out again that with non-uniform stress distribution the stress ratios s may be bigger than $+1$ or smaller than -1 and that, therefore, the limits given for the g value in Table VIII (para. 2,1,2.2) might be exceeded.

Fig. 52



As the theory of plate buckling does not give pertinent information for this case, an approximate solution must be applied. The coefficients of restraint care calculated as if the limiting stress ratios $s = 1$ or $s = -1$ were valid. We therefore introduce a fictitious stress distribution. The procedure is to adopt, for the stress in the plate element for which the s value lies beyond the given limits, an average pure compression stress ($s = +1$) or an average pure bending stress ($s = -1$) and to calculate the stress at the common edge accordingly. With this stress at the edge the stress ratio in the adjacent plate element is now adjusted so that it represents the introduced fictitious stress distribution. The example in Fig. 52 will clarify the procedure.

2,1,3. *THICKNESS t AND WIDTH b OF THE PLATE ELEMENTS*

In para. 2 the author has assumed that the thickness t and the width b are constants.

What is to be considered as thickness t does not require any comments. As far as the width b is concerned, however, it must be defined a little more closely. The cross-section of the shape to be investigated is considered as the net of the centre lines of the individual plate elements. This is permissible as with " blown-up " cross-sections b is considerably bigger than t. Whichever is the actual geometrical form of the edges, *i.e.*, independently whether fillets or toes are present, the edge which is determinant for calculating the width is the intersection of these centre-lines. Changes in thickness due to fillets or toes are not taken into account. Contrary to the above, the actual form of the edge must be taken into account when calculating torsional buckling.

b and t are now considered as variables and how the calculation method can be adapted to this variability can be investigated. We distinguish in this respect the variability which is intentional and the variability which is due to manufacturing tolerances of the construction material.

2,1,3.1. *Intentional Variability*

Variation of b or t might arise when the cross-section of the shape is to be adapted to special conditions. As far as the thickness is concerned, locations where screws or bolts are foreseen are locally increased in thickness. If the wall thickness is to be adapted to the loads, gradual increase of the thickness might be desired ; the variation of the thickness can occur either longitudinally or transversally. In the case of widths, gradual or abrupt change may be necessary when the space available for the structural member is limited or when the variability of b is desired on aesthetic grounds, the designer preferring the conical to the cylindrical form.

All these cases are investigated by calculating the most unfavourable and the most favourable conditions. Depending upon the structural member, the most unfavourable condition must be considered determinant or the designer can, especially when a gradual variability of b or t is present, calculate an average between the most unfavourable and the most favourable condition.

2,1,3.2. *Manufacturing Tolerances*

The influence of manufacturing tolerances upon the critical buckling stress is generally covered by the usual safety factors.

When calculating the coefficients of restraint the ratios $(t_1/t_2)^3$, $(t_1/t_2)^2$ and $(b_1/b_2)^2$ have been used and the entries b_1/b_2 and t_1/t_2 have been used in the diagrams. In the neighbourhood of the straight line $c = \infty$ a small variation of t_1/t_2 or b_1/b_2 produces a very big change in the magnitude of the c values. For the determination of the c values in this region it is necessary to take into account the effects of the manufacturing tolerances when determining the entries. When speaking about manufacturing tolerances it is necessary to distinguish between the various types of semi-finished products.

With structures made from folded sheet, t_1 and t_2 are identical as the two plate elements are made from the same sheet. The value t_1/t_2 need not be corrected. With respect to b_1/b_2 the precision of cutting and of brake pressing the sheet, which varies from manufacturer to manufacturer, must be taken into account. Inasmuch as brake-pressed sheet constructions are used rather for big cross-sections, the errors in b will probably not exceed $\pm 2\frac{1}{2}$ per cent. The ratio b_1/b_2 can, therefore, be erroneous by ± 5 per cent and such an error cannot be neglected in the neighbourhood of the straight line $c = \infty$.

With structures welded-up from sheet the tolerances in the thicknesses for the widths normally used, are about ± 5 per cent. The ratio t_1/t_2 can, therefore, be erroneous by ± 10 per cent, and this error must be taken into account. As the widths depend on the precision of cutting operations only, the error in ratio of widths will be about $\pm 2\frac{1}{2}$ per cent. This error need not be taken into account.

With structures made from extrusions the tolerances in the thicknesses and the widths of the plate elements are about ± 2.5 per cent. The errors in t_1/t_2 and b_1/b_2 are, therefore, ± 5 per cent. These errors must be taken into account.

With hollow extrusions the tolerances in the thicknesses are about ± 10 per cent. The error in the ratio of thicknesses is, therefore, ± 20 per cent. The tolerances in the widths are of the order of ± 5 per cent. The ratio of width, therefore, can be erroneous to the extent of ± 10 per cent. Both these errors naturally must be taken into account.

Extruded tubes or sections which are re-drawn are true in shape so that no errors have to be taken into account.

The above is intended to direct the designer's attention to the fact that if he wants to obtain a perfect structure he has to ask the supplier of the semi-finished products for the manufacturing tolerances. With the most unfavourable tolerances he then reads from the c diagrams the corresponding coefficients of restraint.

Whether in addition he takes account of the above tolerances when calculating the fictitious slenderness ratio $k' \cdot b/t$ is his affair. The author considers a total error of ± 10 per cent as included in the normal safety factor but all in excess of this error is taken into account separately when calculating $k' \cdot b/t$.

2,2. Built-up Sections

We are now in a position to calculate integral sections and we shall investigate how sections which are built-up by riveting can be analyzed. With these built-up sections we determine a fictitious cross-section and calculate it as if it were an integral section. This fictitious cross-section is obtained by applying the following rules.

2,2,1. *RULES FOR THE FICTITIOUS CROSS-SECTION*

The first group of rules refers to the procedure how the rivet rows are dealt with.

(i) Rivet rows are considered as edges. This is permissible because the cross-section is not only calculated as a whole in general buckling, but also the individual plate elements are calculated for general buckling. Plate elements are considered in general buckling as bars with rectangular cross-section and with a buckling length which is equal to the distance of the rivets in the rivet rows. These plate elements, therefore, will not buckle individually and, therefore, the plate must remain straight in the rivet row. With respect to local buckling the rivet row has, therefore, the same properties as an edge. This is all the more the case as here double the material thickness is provided for.

(ii) Double or multiple rivet rows are considered as one fictitious rivet row which passes through the centre of gravity of the rivets.

(iii) Considering that the connexion with a single rivet row is not very stiff it is assumed that the coefficient of restraint is five times as big as for real edges. With double or multiple rivet rows however, the same coefficients of restraint are used as with real edges.

These assumptions give the following approximate k' values for flanges and webs with the usual connexions :—

(*a*) Web encastré between heavy plate elements, $k' = 1.35$ (1.48)* ;

(*b*) Web fixed with rivet rows to light plate elements, $k' = 1.50$ (1.44);

(*c*) Flange connected to heavy plate elements, $k' = 3$ to 3.5 (3.50);

(*d*) Flange connected to light plate elements, $k' = 4$ to 4.5 (4.58).

The values obtained in the calculation example 1 of para. 2,2,2 are given above in parentheses. The value with an * does not agree very well with the approximate solution as the restraining parts are relatively light and the assumption, therefore, not fulfilled. This gives an indication that only the experienced designer should use these approximate figures.

The second group of rules deals with permissible simplifications of the built-up cross-section.

(iv) Ends of plate elements projecting over the rivet rows are not taken into account in the calculation. These ends may buckle locally without jeopardizing the safety of the section as a whole. The application of this rule simplifies the calculation and it may be used also with integral sections where small fins are just ignored in the calculation.

(v) Plate elements in multiple layers are considered as integral plate elements as these multiple layers are not only riveted together along the edges but over their entire surface.

The third group of rules refers to the calculation procedure.

(vi) The width of plate elements which are encastré between other plate elements is taken as the clear unsupported width between these latter plate elements if the former is restrained. If it is restraining the width is calculated in the case of a single rivet row from this rivet row and in the case of multiple rivet rows from the fictitious rivet row.

(vii) The width of plate elements, the edges of which touch only on one side adjacent plate elements, is calculated from the extreme rivet row if the plate element is restrained. If it is a restraining plate element the width is calculated in the case of multiple rivet rows from the fictitious rivet row, with simple rivet rows the procedure is the same as in the case of restrained plate elements.

(viii) It may happen with built-up sections that the limits of the ratio of widths as indicated in Figs. 7 to 10 (paras. 2,1,2.1,1,1-1,1,2-1,1,3-1,1,4) are exceeded. In this case instead of the actual width b an active width b' is calculated so that the limit of the ratio of widths is respected. It is, therefore, assumed that the portion $b - b'$ of the plate element is not active. The critical buckling stress β_k' so obtained applies only to the width b', and the determinant critical buckling stress for the entire width b is calculated with the following formula:—

$$\beta_k = \beta_k' \cdot b'/b$$

Rules (vi) and (vii) are justified because, depending upon the loading conditions, the cross-sections

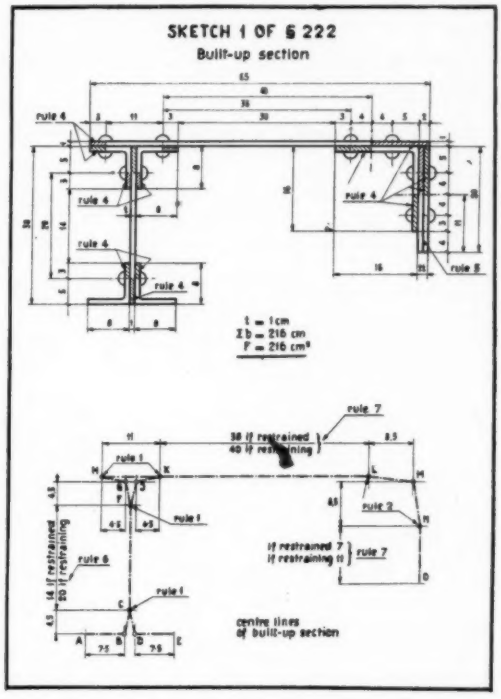


Fig. 53

behave differently and also because the dimensions of the buckles are not infinitely small. Rule (viii) has been drawn up to prevent the designer from adopting unreasonable shapes for "blown-up" cross-sections in aluminium. When applying rule (viii) to plate elements with non-uniform stress distribution, it should not be forgotten that when changing the width b into b' the stress ratio changes.

2.2,2. *EXAMPLE*

The application of these rules will be shown in an example. The author has chosen the cross-section given in Fig. 53, and it is assumed in this example that uniform stress distribution is present. The theoretical buckling formulae of the material are assumed to be as follows —

For $l_k/i < 20.2$: $\beta_k = 2{,}500$ kg. per sq. cm.

For $20.2 < l_k/i < 85.1$: $\beta_k = 2{,}970 - 23.27 \cdot l_k/i$ in kg. per sq. cm.

For $l_k/i > 85.1$: $\beta_k = 7{,}170{,}000/(l_k/i)^2$ in kg. per sq. cm.

EXAMPLE 1 *OF PARA.* 2,2,2

Built-up Section

Edge B :

flange *AB* is restrained

$b_1 = 7.5$ cm. $t_1 = 1$ cm. (flange)

$b_2 = 4.5$ cm. $t_2 = 1$ cm. (web)
$b_1/b_2 = 7.5/4.5 = 1.67$; $t_1/t_2 = 1$
but from Fig. 8 (para. 2,1,2.1,1,2) and rule 8 then
$b'_1 = 4.5 \times 1.25 = 5.62$ cm.
from Fig. 8 (para. 2,1,2.1,1,2): $c = 0.70$
from Fig. 4 or 5 (para. 1,2,1): $k' = 3.50$
$(l_k/i) = 3.5 \times 5.62/1 = 19.6$
$\beta_k' = 2500$ kg./cm.²
$\beta_k = 2500 \times 5.6/7.5 = 1875$ kg./cm.²

Edge D :
flange *DE* is restrained, same values as for edge *B*

Edge C :
web *CF* is restrained.
$b_1 = 14$ cm. $t_1 = 1$ cm. (web)
$b_2 = 4.5$ cm. $t_2 = 1$ cm. (web)
$b_3 = 4.5$ cm. $t_3 = 1$ cm. (web)
$b_1/b_2 = 14/4.5 = 3.11$; $t_1/t_2 = 1$
from Fig. 10 (para. 2,1,2.1,1,4): $c'' = 0.27$
half value since there are two restraining webs:
$c' = 0.27/2 = 0.135$
by reason of rule 3 we obtain: $c = 5.\ 0.135 = 0.675$
from Fig. 1 (para. 1,1,2): $k' = 1.48$

Edge F :
web *CF* is restrained, same value as for edge *C*
$(l_k/i) = 1.48 \times 14/1 = 20.7$
$\beta_k = 2970 - 23.27 \times 20.7 = 2488$ kg./cm.²

Edge G :
web *FG* and *GH* have identical dimensions
from Fig. 10 (para. 2,1,2.1,1,4) we get c = ∞, the webs do not interact.
From the examination of edge *F* it follows that they are restraining.

Edge J :
as for edge *G*.

Edge H :
web *HK* is restrained.
$b_1 = 11$ cm. $t_1 = 1$ cm. (web)
$b_2 = 4.5$ cm. $t_2 = 1$ cm. (web)
$b_1/b_2 = 11/4.5 = 2.45$; $t_1/t_2 = 1$
from Fig. 10 (para. 2,1,2.1,1,4): $c' = 0.34$
by reason of rule 3 we obtain $c = 5 \times 0.34 = 1.7$
from Fig. 1 (para. 1,2,1): $k' = 1{,}56$
$(l_k/i) = 1.56 \times 11/1 = 17.1$
$\beta_k = 2500$ kg./cm.²

Edge K :
web *KL* is restrained, web *HK* co-restraining
$b_1 = 36$ cm. $t_1 = 1$ cm. (web)
$b_2 = 4.5$ cm. $t_2 = 1$ cm. (web)
$b_3 = 11$ cm. $t_3 = 1$ cm. (web)
$b_1/b_2 = 36/4.5 = 8.0$ $t_1/t_2 = 1$
$b_1/b_3 = 36/11 = 3.28$ $t_1/t_3 = 1$
but from Fig. 10 (para. 2,1,2.1,1,4) and rule 8:
$b'_1 = 4.5 \times 5 = 22.5$ cm.
therefore $b'_1/b_3 = 22.5/11 = 2.05$
from Fig. 10 (para. 2,1,2.1,1,4): $c'_{12} = 0.2$
$c'_{13} = 0.4$
by reason of rule 3 $c_{12} = 5 \times 0.2 = 1.0$
$c_{13} = 5 \times 0.4 = 2.0$
from para. 2,1,2.1,2,1 $1/c = 1/1+1/2 = 1.5$;
$c = 0.67$
from Fig. 1 (para. 1,2,1); $k'_a = 1.48$ and $(l_k/i)'_a = 1.48 \times 22.5/1 = 33.3$
$\beta_{ka} = [2970 - 23.37 \times 33.3] \cdot 22.5/36 = 2195 \times 22.5/36 = 1372$ kg./cm.²

Edge L :
web *KL* is restrained
$b_1 = 36$ cm. $t_1 = 1$ cm. (web)
$b_2 = 8.5$ cm. $t_2 = 1$ cm. (web)
$b_1/b_2 = 36/8.5 = 4.25$; $t_1/t_2 = 1$
from Fig. 10 (para. 2,1,2.1,1,4): $c = 0.23$
from Fig. 1 (para. 1,2,1): $k'_b = 1.39$
and $(l_k/i)_b = 1.39 \times 36/1 = 50.0$
$\beta_{kb} = [2970 - 23.27 \times 50] = 1805$ kg./cm.²
$\beta_k = (\beta_{ka} + \beta_{kb})/2 = (1372 + 1806)/2 = 1589$ kg./cm²
We can now finish the calculation for web *HK* according to para. 2,1,2.1,2,4
$\beta_k = (2500 + 1589)/2 = 2045$ kg./cm.²

Edge M :
webs *LM* and *MN* have identical dimensions, they do not interact and are restraining.

Edge N :
flange *NO* is restrained
$b_1 = 7$ cm. $t_1 = 2$ cm. (flange)
$b_2 = 8.5$ cm. $t_2 = 1$ cm. (web)
$b_1/b_2 = 7/8.5 = 0.825$; $t_1/t_2 = 2/1 = 2$
from Fig. 8 (para. 2,1,2.1,1,2): $c = 22$
from Figs. 4 or 5 (para. 1,2,1): $k' = 4.58$
$(l_k/i) = 4.58 \times 7/2 = 16$
$\beta_k = 2500$ kg./cm.²

The web *KL* has the smallest critical buckling stress, for the local buckling the value $\beta_k = 1589$ kg./cm.² therefore becomes determinant. The total critical buckling load is as follows:—

$P = 1589 \times 216 = 343{,}200$ kg.

In the calculation of example 1 (para. 2,2,2) the critical buckling load has been favourably calculated in assuming that the ends of the plate elements protruding over the rivet rows have at least a local buckling stress of 1589 kg. per sq. cm.

Formally, this calculation example is finalized, but in practice one would try to improve the chosen cross-section within the overall dimensions which are supposed here to be invariable. In fact, it has been demonstrated with this calculation that six plate elements buckle at various critical buckling stresses.

Flanges AB and DE	$\beta_k = 1875$	kg. per	sq. cm.
Web CF	$\beta_k = 2488$	,,	,,
Web HK	$\beta_k = 2045$	,,	,,
Web KL	$\beta_k = 1589$	,,	,,
Flange NO	$\beta_k = 2500$	,,	,,

The conditions at flanges AB and DE can be improved by providing for a cover plate AE. This new cover plate AE will buckle and not the flanges AB and DE. The thickness of this new

cover plate can be so chosen that $\beta_k = 2500$ kg. per sq. cm².

The conditions at webs HK and KL can be improved by providing for a cover plate HKL and by doubling this cover plate by a second one over the distance KL. In doing so it is possible to improve β_k for both webs HK and KL so that local buckling occurs at the yield point.

If the general buckling of the section investigated does not occur at the yield point but at a lower stress, the local buckling stress should be adapted to this lower figure, in doing so the thicknesses of all the plate elements in the example can be reduced correspondingly.

2,2,3. *COMPARISON WITH AN INTEGRAL SECTION OF THE SAME WEIGHT*

In order to compare the strength of built-up and integral sections, an integral section is calculated with the same overall dimensions and the same weight as the section given in Fig. 53 (para. 2,2,2). This calculation shows how much more simply and rapidly integral sections can be dealt with.

EXAMPLE 1 OF PARA. 2,2,3

Integral Section

Edge B :

flange *AB* is restrained

$b_1 = 16.19$ cm. $t_1 = 1.62$ cm. (flange)

$b_2 = 29.38$ cm. $t_2 = 1.62$ cm. (web)

$b_1/b_2 = 16.19/29.38 = 0.55$, $t_1/t_2 = 1$

from Fig. 8 (para. 2,1,2.1,1,2) : $c = 1.3$

from Figs. 4 or 5 (para. 1,2,1) : $k' = 3.65$

$(l_k/i) = 3.65 \times 16.19/1.62 = 36.4$

Edge C :

web *CD* is restrained

$b_1 = 47.19$ cm. $t_1 = 1.62$ cm. (web)

$b_2 = 29.38$ cm. $t_2 = 1.62$ cm. (web)

$b_1/b_2 = 47.19/29.38 = 1.60$, $t_1/t_2 = 1$

from Fig. 10 (para. 2,1,2.1,1,4) : $c_c = 0.50$

from Fig. 1 (para. 1,2,1) : $k'_c = 1.46$

Edge D :

web *CD* is restrained

$b_1 = 47.19$ cm. $t_1 = 1.62$ cm. (web)

$b_2 = 20.19$ cm. $t_2 = 3.24$ cm. (flange)

$b_1/b_2 = 47.19/20.19 = 2.34$, $t_1/t_2 = 0.5$

from Fig. 9 (para. 2,1,2.1,1,3) : $c_d = 0.05$

from Fig. 1 (para. 1,2,1) : $k'_d = 1.30$

the nomogram Fig. 6 (para. 1,2,2) gives with k'_c and k'_d :

$k' = 1.38$

$(l_k/i) = 1.38 \times 47.19/1.62 = 40.2$

The greatest fictitious slenderness ratio is, therefore, 40.2.

The corresponding critical buckling stress is :

$\beta_k = 2970 - 23.27 \times 40.2 = 2035$ kg./cm.²

and the total critical buckling load becomes :

$P = 2035 \times 216 = 439{,}000$ kg.

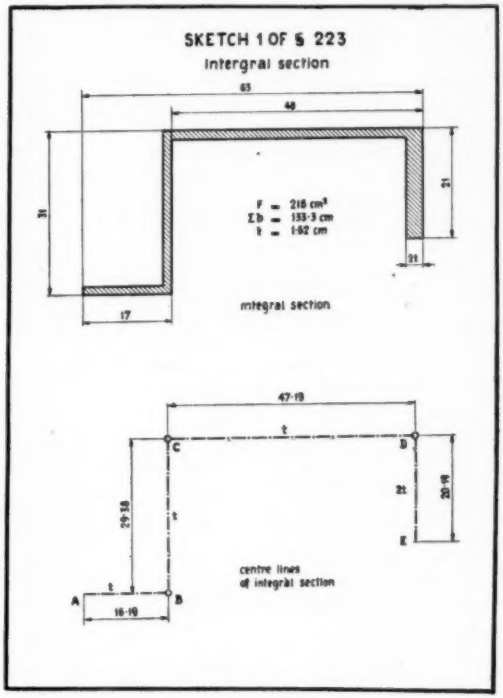


Fig. 54

Compared with the built-up section the load bearing capacity is :

439,000/343,200 = 1.28 times greater.

3. PERMISSIBLE STRESSES

The investigation of local buckling has been limited in general up to now to the calculation of fictitious slenderness ratios. In a few cases, however, it was necessary to calculate the critical buckling stresses in order to obtain an approximate solution for a particular case. From these critical buckling stresses the permissible stresses are obtained by dividing with the factors of safety. Critical stresses and factors of safety will be discussed hereafter, more in detail.

3,1. *The Critical Buckling Stresses*

The critical buckling stresses as already said in para. 1,2,1 should be taken from a theoretical buckling diagram. This means that these buckling stresses are established for a mathematically straight member and that they represent the highest possible buckling stresses which may be obtained with a given material. The formulae given in para. 2,2,2 meet these requirements.

It is superfluous to base the calculation of the critical buckling stress in local buckling upon a practical buckling diagram which takes into account

the influence of unavoidable eccentricities. According to the author's definition, local buckling occurs before the compression member buckles as a whole therefore eccentricities which are present have an influence only upon the stress distribution in the cross-section and this has already been accounted for in the statical calculation.

The particular conditions prevailing with local buckling in respect of imperfect fulfilment of the assumptions have been considered when calculating the ratio of widths and ratio of thicknesses. The reduction of the theoretical buckling diagram into a practical buckling diagram has thus already been accomplished. It is an unjustifiable waste of material if the fictitious slenderness ratios are introduced into a practical buckling diagram for eccentric buckling.

3.2. *Factors of Safety*

The choice of the factors of safety is made according to the following rules. If local buckling limits the strength of the construction member or if it causes its failure the safety factor must be as follows :—

$$n = 2.5$$

If through local buckling the strength of the construction member is not limited then the safety factor is :—

$$n = 2$$

It is supposed here that the buckles will not be visible and, therefore, the appearance of the member is not jeopardized. In the case that visible buckles are acceptable, the safety factor may be

$$n = 1.5$$

In this case the appearance of the construction is slightly marred.

These conditions will be explained more in detail with an example. In an equal angle under uniform stresses the critical local buckling load of the legs limits the strength of the section if the stress lies in the elastic range, and it means the failure of the member if the stress is in the plastic or semi-plastic range. In this case, therefore, $n = 2.5$.

In the case of an equal angle under non-uniform stress distribution or in the case of unequal angle under any type of stress distribution, reaching the critical local buckling stress in one leg does not mean that the final strength of the section is attained. The other leg is capable of taking on still more load until in turn it buckles locally. If visible buckles must be avoided the safety factor should be $n = 2$. If, on the contrary, visible buckles are not objectionable the safety factor may be $n = 1.5$.

Very generally, therefore, it is possible to say that in a section where all the plate elements buckle simultaneously, $n = 2.5$, if only one of several plate elements buckles, $n = 2$ or $n = 1.5$ depending whether the buckles should remain invisible or whether they can become visible.

4. CONCLUSIONS

The author has now given all the information a designer should possess about local buckling of plate elements. This study is specially addressed to designers. We have not followed any scientific ambitions as can be seen clearly from the way in which the subject has been treated. The intention is solely to communicate the experience made during 30 years in the design office of the author's organization and the methods of calculation derived therefrom.

The size of the study may induce the designer not to tackle the problem of local buckling as under pressure of daily work he can hardly afford to carry out lengthy calculations. We should, therefore, like to emphasize in particular that these calculations are not very time-consuming and that the length of our study is solely due to the explanations of the diagrams given.

What the designer requires is :—

(i) The nomogram of para 2,1,2.2 (Fig. 15) for the determination of the g values ;

(ii) Figs. 16 to 51 (paras. 2,1,2.2,1,-2,2-2,3-2,4) for the calculation of the c values ;—

(iii) Figs. 1 to 5 (para. 1,2,1) for the calculation of the k' values ;

(iv) The nomogram of para. 1,2,2 (Fig. 6) for the average of the k' values.

Herein all the lengthy calculations are already included and the designer has only to learn the use of these diagrams to profit from them when designing " blown-up " cross-sections. If, in addition, the designer respects the limitations contained in the diagrams, he will obtain an acceptable structure which will not deflate forthwith under the critical eye of the specialist.

★ ★ ★

INTRODUCING his paper, Dr. Sutter said that it might appear to be rather on the theoretical side and by theoretical he meant that it did not contain any reference to its practical application.

He, therefore, proposed to show as an introduction to the paper that aluminium alloys are widely used in buildings and other structures in the form of sheet metal ; that the structurally more elaborate of these applications require calculation ; and, finally, that such calculations lead, with sheet-metal structures, directly to the subject of his paper.

He would show a number of colour slides of jobs with which Aluminium Laboratories Ltd., Geneva, were connected ; the slides were segregated into two main groups : Buildings and Structures, and referred to the following :—

Buildings

Roofing : Flat sheet ; corrugated sheet ; pressed sheet.

Sidewall Cladding (plain and coloured) : Flat sheet ; corrugated sheet ; pressed sheet.

Floors : Self-supporting elements.

Miscellaneous : Shadelight canopies ; louvres ; eaves ; railings.

Structures

Tubular Sections made from Sheet : Flagpoles ; transmission line masts ; lighting towers.

Open sections made from sheet : Self-supporting troughs for steam pipes ; stiffening beams for suspended footbridges.

Structures using tubular and open sections : Transmission line towers ; cross beams and tubular legs ; footbridges ; girders and tubular pillars ; snow fences ; planks, beams and tubular struts.

The technicalities of the paper would be dealt with after the discussion to such an extent as is deemed necessary.

DISCUSSION

Mr. I. R. BANKS (Aluminium Laboratories Ltd.) complimented Dr. Sutter on what was really a monumentally thorough analysis of the phenomenon of local buckling, not necessarily in aluminium. He looked upon it as a background document for codes of practice to come, and it was very fortunate for all concerned with this problem to have it written into the Proceedings of the Institute. He was also to be complimented on having given examples of how to use aluminium, because there were some people who were not able to follow the intricacies of local buckling without such help.

Coming to criticism, he asked if Dr. Sutter would give the references to the sources or origin in the local quantities that were given in the paper, for there were many who were so far behind him in this work that they would like to have the information more readily available.

Further criticisms, said Mr. Banks, were minor, and one of them, which he had already mentioned to Dr. Sutter, concerned the matter of language. It was not a criticism of Dr. Sutter, for he had presented the paper to a British audience ; who among them would be able to present such a paper in Switzerland ? Some words had different meanings in the two countries, and he asked that those responsible for the editing would see that some of the words were changed.

For example, on the first page of the paper, when referring to the compression member beginning to fail locally before buckling, it was pointed out that the " deformation of the compression member becomes finite instead of infinite . . " The English translation of " infinite " was " very small."

On the second page the coefficient of restraint was regarded as infinite for a simply supported edge. That again was contrary to our thinking ; we would say that the coefficient of restraint was infinite at a fixed edge.

The work in the paper was entirely theoretical, and in that respect the paper was in contrast with the papers presented at the previous session. It led Mr. Banks to believe that there would be room for a certain amount, he did not quite know how much, of experimental work to prove some of the points, under Dr. Sutter's supervision. Four sets of coefficients were given, which set was nearest to practical work ? And was more research work needed ? He was sure that in the aluminium industry of this country there was someone who was prepared to sponsor the research that Dr. Sutter considered to be necessary.

Replying to Mr. Banks, Dr. SUTTER said that the question regarding a more detailed bibliography to the paper would best be answered as follows :—

Detailed sources of information relating to paragraph 1.1 : " Generally known basis of calculation " were not given as in the various countries the relevant sources are different and, therefore, only of local interest. The coefficients k for plate buckling can be found in any textbook.

The representation of these coefficients in diagrams Fig. 1 to 5 is the result of the author's own work. The two border curves and the values for the stress ratio $s = 1$ have been taken from technical literature. The calculation of the inside curves, however, was done by Aluminium Laboratories, and the procedure of F. Bleich was used for this calculation.

The coefficient k is, according to F. Bleich, page 324, " Buckling strength of metal structures " :

$$k = p + 2\sqrt{q}$$

where, in the case of flanges (page 328)

$$p = 0.425 + 0.016/(0.025 + c)$$

$$q = 0.061/(0.43 + c)$$

and in the case of webs (page 323)

$$p = 2 + 0.047/(0.73 + c)$$

$$q = 1 + 0.70/(0.077 + c)$$

c is the coefficient of restraint, and the values obtained for flanges have been corrected as shown in Table II of the paper to meet the conditions true for aluminium.

Coefficient of Restraint

The diagrams for determining the coefficient of restraint, Figs. 7 to 10 and Figs. 16 to 51 have been calculated by the author using the 4 formulae indicated in the paper. The values of the functions f_s and f_F which appear in these formulae were taken from F. Bleich, pages 335 and 338, and partly calculated by the author, the field of the diagrams being wider than Bleich's diagrams.

For restraining flanges the following is valid :

$$f_F = \frac{3\cosh^2[\pi b_F/(db)] + [\pi b_F^2/(db)]^2 + 1}{[2\pi b_F/(db)]\{\pi b_F/(db)\, 3\sinh[\pi b_F/(db)].\cosh[\pi b_F/(db)]\}}$$

where b_F = width of the restraining flange

db = half wave length of the restrained plate

For restraining webs

$$f_s = \frac{\tanh[\pi b_s/(2db)]\{1+[\pi b_s/)db)]/\sinh[\pi b_s/(db)]\}}{\pi\, b_s/(db)}$$

where b_s = width of the restraining web

db = Half wave length of the restrained plate.

The actual calculation of the diagrams was carried out in the following way :

A $b_s/(db)$ or $b_F/(db)$ value was assumed and the function f_F or f_s calculated, then a c-value was assumed and the corresponding half wave length determined.

$$d\,.\,b = 4\sqrt{1/q}$$

q being a function of c as indicated before. At this juncture the value $b_s/(db)$ can be expressed by the ratio of widths b_s/b_F, etc., as used in the diagrams. Now a ratio of thickness t_s/t_F is assumed and the $t/_s\ t_F$ value calculated according to one of the four formulae for the c-values. By trial and error t_s/t_F was so adjusted that the calculated t_s/t_F value coincided with the assumed t_s/t_F value and was then plotted in the diagram.

The establishment of the 36 diagrams with hundreds of c-curves represented a considerable amount of work but this can be justified by the very general design rules thus obtained. Incidentally, F. Bleich derives from this theory only approximate design rules for standard steel sections (page 346); the author's aim was to produce a method which permits the evaluation of local buckling for any type of section. F. Bleich does not give any diagrams for the coefficients of restraint of the type the author has established.

The rest of the paper is a write-up of experience with the use of this calculating method and its adaptation to special conditions encountered in practice. Although no basic tests have been made to establish the method, some of the structures designed in accordance with the method were tested and it was found that there was no relevant discrepancy between the failing loads predicted and the actual failing loads obtained in tests. Some of the tests were shown in the colour slides presented earlier in this meeting.

The question, which one of the 4 sets of curves is to be used when analysing a given section, he would answer as follows :—

In the case of simple sections as shown in Figs. 7 to 10, the designer can easily find out to which group his section belongs. In the case of more complicated sections, he picks out the plate element he thinks is buckling, this plate element is either a flange or a web, for flanges Fig. 7 and 8 apply while for webs Figs. 9 and 10 apply. Assuming that the plate element is a web, then the designer determines whether the web is restrained by flanges or by webs. If flanges are the restraining elements, Fig. 9 applies; if webs are the restraining elements, Fig. 10 applies. If the web is restrained at each edge by more than one plate element, he applies the rules given in paragraphs 2,2,1.1,2,1 to 2,2,1.1,2,4. As long as the designer finds in the c diagrams a value for the coefficients of restraint, his assumption which plate element is buckling is correct. If he does not find a c value, this means that another plate element than the assumed one is buckling and he must change his assumption until he can read from the diagrams all the necessary c values. In the author's experience, even inexperienced designers find very quickly all the necessary c values. Such trial and error solutions are very frequently used in solving difficult engineering problems and no objections had been heard of to this aspect of the method.

Buckling of Channel Sections

Mr. J. B. DWIGHT (Birmingham University) mentioned an aspect of local buckling which rather worried him. With a channel section it was possible to work out the local buckling strength with the aid of the comprehensive data provided by Dr. Sutter or the authors of former papers, including that by Dr. Chilver.

Illustrating some channel sections, with one with very wide lips it was possible, with the aid of Dr. Sutter's paper, to work out the local buckling strength, assuming that the section buckled in such a way that the four junctions remained straight. But in his experience of design one tended to use sections in compression which had relatively small lips, and the problem was, how large must the lips be to make it possible to work out the local buckling strength of such a section? Also, if the lips were small enough to enable the section to behave like an unlipped channel, how was it allowed for? He added that in the aircraft field there was a paper which dealt with work started in about the year 1946, but he had not seen any follow-up.

He suspected that the steel people, in order to keep their designs as simple as they had done, had preferred to disregard the small lip and to call it either a flange or a web, and not to have any half-way stage.

Dr. SUTTER said that he had dealt in paragraph 2,1,2.1,3 only very briefly with the efficiency of lips and had not given a direct answer to Mr. Dwight's question. This answer could, however, be found in Aluminium Development Association Research Report No. 29 mentioned in the paper.

Indirectly, an approximate answer could be found for instance for uniform stress distribution and lips of the same thickness as the restrained plate elements in Fig. 9. For a ratio of thickness $t_1/t_2 = 1$ and $c = \infty$, *i.e.*, no restraint, $b_1/b_2 = 3.1$ or $b_2/b_1 = 0.32$. For values above 0.32 the lips would, therefore, buckle. b_2 is the width of the lip and b_1 the width of the restrained plate element.

On the other hand, the ratios b_1/b_2 were limited in this figure to the value of 10 and for values above 10 the diagram could not be used; therefore the smallest practical lip had a ratio of $b_2/b_1 = 0.1$ and produced a coefficient of restraint of approximately $c = 0.7$ upon the adjoining plate element.

Also, from Fig. 9, the smallest coefficient of restraint occurred at a b_1/b_2 value of approximately 6, therefore the ideal width of the lip had a ratio $b_2/b_1 = 0.17$. The corresponding c value was approximately 0.6.

It might appear strange that he had hidden in the paper the treatment of lips, as lips were admittedly a very powerful means to increase the efficiency of a section. However, he would point out again that he had adopted, when dealing with local buckling, the viewpoint of the designer familiar with ordinary aluminium alloy structures who tried to obtain a more economic design by inflating the cross-sections of the members.

The first step in inflating the cross-section was to make its wall thickness thinner and its overall dimensions bigger. This first step might already be sufficient to obtain a competitive structure. As a second step, lips could be added to this inflated section and a still lighter structure thus obtained. However, adding lips meant in most cases design trouble at connexions; the savings in metal weight could easily be counterbalanced by increased cost of manufacture. For these economical reasons, he did not emphasize the mechanical merits of lips.

Buckling at Lips

Dr. A. H. CHILVER (Dept. of Engineering, Cambridge University) commented on Dr. Sutter's paper and Mr. Dwight's remarks about sections buckling at the lips.

In the paper Dr. Sutter advocated design on the basis of collapse or buckling. Dr. Chilver considered that full use was not being made of the properties of the material for specifications involving initial buckling stresses rather than collapse stresses.

Towards the end of paragraphs 1,1,1. of his paper, Dr. Sutter had stated that with T or cruciform sections and the like the centre of shear coincided with the one edge on which all the plate elements joined and the warping constant was zero. They were, therefore, prone to fail as a whole in torsional buckling, and torsional buckling was in this case more unfavourable than local buckling. It seemed to Dr. Chilver that in this particular case the local and torsional buckling were indistinguishable.

Again in paragraph 1,1. the author appeared to introduce a generalization. Towards the end of page 1 he stated: "With this transformation of the equation β_k it is now possible to calculate the critical buckling stresses for local buckling with the Euler formula or they can be read with the entry (l_k/i) from a theoretical buckling diagram. When doing so not only is the elastic behaviour taken into account—upon which the Euler formula is based—but also the inelastic behaviour of the material." Dr. Chilver could not quite follow that: however, he had not had a chance to study it in detail.

A point to remember was that interaction between component plates in a simple design was possible only when the wavelength of buckling in two adjacent plates was of the same order. It seemed to him to be rather rash to suggest the introduction of simple interaction between the flanges and web of a beam.

On the last page of the paper Dr. Sutter made a comment about eccentricities and seemed to suggest that they were not very serious. He stated: "According to the author's definition local buckling occurs before the compression member buckles as a whole, therefore, eccentricities which are present have an influence only upon the stress distribution in the cross-section . . ." Dr. Chilver did not think that one could ignore the initial deflections.

Finally, on the point of local buckling, he illustrated the behaviour of a single plate with a rib on one edge, and the effects of stress. The problem, he said, had been solved for the symmetrical case, but not for the unsymmetrical case.

Dr. SUTTER, answering Dr. Chilver, said that the remarks and questions which came up in the discussion as well as the comparison of his paper with the paper "The Contribution of Research in Developing the Use of Cold-rolled (steel) Sections in Buildings," by R. M. Kenedi and A. H. Chilver, indicated that he should explain here in more detail the approach chosen when dealing with local buckling.

The viewpoint of the designer who is familiar with ordinary aluminium structures had been adopted, *i.e.*, structures or structural elements which did not fail in local buckling. It was then assumed that the designer was compelled to produce a more economic structure by introducing "blown-up" or "inflated" sections, and a procedure of calculating local buckling was shown which the designer could use on any type of section

he might be called upon to introduce in a structure. This meant that the section could not be identified—even approximately—with a section for which extensive research had been done and the exact behaviour of which was thus not known under loading.

Dr. Sutter's paper was only a section out of a much more complete study of buckling phenomena and, therefore, at its fringe; some of the terms used were not defined in detail, because this was done in other parts of the complete study. This in particular applied to the term theoretical buckling diagram. A theoretical buckling diagram represented the buckling stress in relation to the slenderness ratio; it was composed—with increasing slenderness ratios—of the yield point, the Tetmajer straight line and the Euler hyperbola.

Theoretical did not mean that he used over the entire range of slenderness ratios the Euler hyperbola. Such a buckling diagram was designated as an ideal buckling diagram. He used the designation theoretical buckling diagram in opposition to a practical buckling diagram (*e.g.*, Perry-Robertson) to indicate that no accidental eccentricities of the structural member were taken into account.

The paper dealt basically with the determination of the coefficient k' used for calculating fictitious slenderness ratios of plate elements as used in the calculation of local buckling. For curved elements, another set of coefficients k″ was required; this was the subject of another paper.

No information about the theoretical buckling diagrams into which the fictitious slenderness ratios were introduced was given; this subject also was treated in another paper.

Practical buckling diagrams were the subject of still another paper as well as the combination of flexural and torsional buckling and the interaction between general and local buckling.

The limitation of this subject of the paper under discussion was necessary to keep it concise but this did not mean that he was not aware of the buckling phenomena which lay beyond the direct scope of the paper.

Having introduced this calculation method in a number of design offices of Continental enterprises, he had noticed that certain aspects of the calculation of local buckling were regularly misunderstood and he had, therefore, in the paper warned the designers of the more common pitfalls. He had done so in the language the designer understood and this might be interpreted by the scientist, though incorrectly, as making rash statements.

(i) A common error when calculating local buckling stresses was to use formulae containing the modulus of elasticity (*see* para. 1.1.). He therefore insisted that a fictitious slenderness ratio $k'.b/t$ be calculated and introduced in what was called a theoretical buckling diagram. There was no possible doubt that this theoretical buckling diagram was based on the tangent modulus and not upon the elastic modulus of the material, and for the convenience of the designer it was replaced with negligible error by the yield point—the Tetmajer straight line—and the Euler hyperbola, and when designing in aluminium alloys an endeavour was made to keep the slenderness ratios below the ideally elastic range.

This answered Dr. Chilver's remark about his doubts that using a fictitious slenderness ratio as entry into the theoretical buckling diagrams would make it possible to take into account the non-elastic behaviour of the material.

(ii) The designers, not having the scientific background of a research team, were generally confused by the relevant literature on the interaction of local buckling and general torsional buckling. This interaction certainly existed but no designer could figure it out in the short time he was usually given for preparing a project.

Dr. Sutter, therefore, kept the two phenomena separate by adopting an appropriate design, and local buckling meant the crippling of one plate element constituting a section in a number of buckles while torsional general buckling meant the failure through twisting of the section as a whole in one half wave length.

As shown in the paper, he used for calculating the critical load of a compressive member its total area and the critical buckling load of the *weakest* plate element. Thus the use of the full strength of the section was neglected as some of the restrained plate elements had critical stresses above that of the weakest plate elements and some others were restraining plate elements.

The inflating process of the section stopped when *one* plate element buckled locally; what happened beyond this point was neglected for the sake of the simplicity of the design rules. Proceeding in this way avoided entering the region of uncertain results where local and general buckling interacted noticeably.

Interaction of Local and General Buckling

This answered Dr. Chilver's remark about the interaction of local and general buckling. It also answered his statement that adjoining plate elements buckled with different wave lengths—the loads were pushed only up to a point where *one* plate element buckled; what happens beyond this point was not suitable to be included in designers' rules.

In paragraph 1.1.1. Dr. Sutter said he made the remark that the warping constant was zero for some types of sections. It was evident that this referred to the primary warping constants as there

was no need, for practical purposes, for the designer to calculate the existing but negligible secondary warping constant.

(iii) The designer would naturally ask himself about the consequences of the simplifications. The error introduced was less than if he introduced in his calculations two types of elements only, *i.e.*, " stiffened elements " and " unstiffened elements " (*i.e.*, neglecting the coefficients of restraint) or if he calculated the local buckling for uniform stress distribution only, neglecting the influence of non-uniform stress distribution (*i.e.*, taking the stress distribution factor unity).

(iv) Having calculated the fictitious slenderness ratio for local buckling, the designer would query which type of buckling diagram he had to use to get the critical stresses. As pointed out in item (i) the theoretical buckling diagrams were used.

It was possible, however, to use diagrams for crippling or collapse stresses if these were available. In the author's experience, however, the designers had only on rare occasions access to such diagrams, remembering that there were hundreds of aluminium alloys on the market all over the world; calculations were therefore based on the more conservative buckling diagrams.

This answered Dr. Chilver's remarks regarding the desirability of using diagrams for crippling stresses.

The Perry-Robertson Formula

(v) In some instances Dr. Sutter had noticed that designers introduced the fictitious slenderness ratios for local buckling into practical buckling diagrams of the Perry-Robertson type. These gave much lower critical stresses and this was an unnecessary precaution when calculating local buckling stresses.

The Perry-Robertson formula gave the influence of unavoidable eccentricity upon general flexural buckling stresses. When calculating the stress distribution in a cross-section, Dr. Sutter, however, took into account the influence of the specified eccentricities upon the stress distribution used in the calculation of local buckling. The influence of these eccentricities upon the general flexural buckling stresses was neglected and the section was designed in such a way that local buckling occurred just before general buckling set in. The deflections at which general buckling set in with elastic-plastic materials were not nil but they were still negligible.

This answered Dr. Chilver's remarks regarding the way of taking into account in the calculations unavoidable eccentricities.

(vi) Designers were also very often confused by the multitude of procedures which had been recommended for dealing with local buckling. As could be seen from the foregoing, it was considered that the buckling stress of the most unfavourable restrained plate elements was the determinant.

Higher critical loads were obtained when the notion of active width of the plate elements was introduced. In the case of very wide and very thin sheet, in the state of collapse, only a part of the element was carrying the load, the calculation was then carried out with a reduced fictitious cross-sectional area. Procedures which calculated reduced crippling stresses for the various plate elements were related to the procedure of active width. Instead of reducing the area, the stresses were reduced; the final result was similar. With these procedures the deformation of certain parts of the sections became excessive and were often permanent, a state of affairs the designer in aluminium alloy tried to avoid.

He had used in the paper the notion of active width only to be able to use in exceptional cases the diagrams of the coefficient of restraint beyond the recommended ratios of widths or thicknesses (*see* paragraph 2.2.1).

(vii) After having calculated the critical load in local buckling a member could carry, most designers considered their job done. This, however, was not so. The method of dealing with local buckling gave him a tool to improve the originally-assumed shape of the cross-section so that it carried a maximum of load with a minimum material required. Dr. Sutter's method showed which plate element was the weakest in a given section and supplied the designer with an indication where to improve the chosen cross-section. It was conceivable that for standard sections the shape of the best cross-section could be indicated in general terms, but unfortunately the designer who used aluminium alloys had to deal very often with sections which defied any standardization and he therefore needed a tool to deal with them efficiently. Dr. Sutter did not believe the basic research would ever be able to produce such a universal tool.

Replying to a question by Mr. GREEN (Tubewrights Ltd.), Dr. Sutter said that the aeronautical engineers had developed efficient means for calculating aluminium alloy structures and that the structural engineer should look upon them for guidance, as Mr. Green said.

In practice, however, the application of this idea met with some snags. Only in very exceptional cases was the information the aeronautical engineers had made available to the structural engineer, therefore the latter must develop his own methods to solve his problems. This was a most regrettable duplication of effort.

(*Continued in page* 62)

Local Buckling of Aluminium Plate Elements

(*Continued from page* 61)

Another and more serious trouble was that the standards in calculation and workmanship were so much more exacting in aeronautical engineering than in structural engineering that these standards must be relaxed to obtain competitive solutions. In structural engineering what this meant was best shown in an example :

The cost of ordinary aluminium structures in Switzerland was as follows :—

	Fr/kg.
Material (on weight of main parts only) approx. cost .. .. .. =	5.–
Labour (including cost of minor parts) approx. cost .. .. .. =	10.–
	15.–

The first comment on these figures was that labour costs more than material ; in order to obtain competitive structures, therefore, the labour cost must be kept down and it was necessary not to worry too much about material cost or extreme lightness of the structure. This excluded in structures all the weight-saving " tricks " obtained through a complication of the design and the ensuing increase of labour cost. What was acceptable were weight savings due to more exact calculation but which were not coupled with a refinement of the structure which was not common in structural engineering.

Only on very rare occasions had the author heard of structures built in accordance with the principles of aeronautical engineering ; all such structures cost a multiple of the figure quoted above.

In transport engineering it was more usual to encounter structures built in accordance with the principles of aeronautical engineering. He remembered a case where self-supported folding sheet-steel bus bodies were built in this way. But also here the costs were 30 per cent higher than similar bus bodies designed along conventional lines and the weight only slightly less, so that the savings in operation paid off the increased first cost only after several years.

In summarizing the situation, he could say that the aeronautical engineer must forget a great deal of his experience until he could turn out competitive designs in the fields of structural and transport engineering.

As regards rounded-off corners of sheet-metal sections, the structural designer liked them sharp because they left more clearance for riveting and fastenings. He had, therefore, assumed in the investigation sharp corners, which gave the greatest width of the plate elements and hence represented the most unfavourable conditions. The k' values are calculated for this condition.

He had found that when a designer obtained too low a critical stress he was inclined to interpose curved parts between the plate elements, thus reducing the width of the plate elements and consequently obtaining a higher critical stress in local buckling. Such edges, however, were not as stiff as sharp edges and the calculated gain in critical stress was not achieved in practice. In order to stop such a fallacious procedure he insisted that the width of any plate element be calculated from the fictitious point of intersection of the middle lines of the adjacent plate elements (*see* paragraph 2.1.3).

The Ariel " Leader "

(*Continued from page* 22)

the rear end is assembled including the pivoted fork, rear wheel, chain drive, chain case, etc.

The wiring harness is then connected up and the anti-theft lock and front wheel added. The next stage is to assemble the body shell, handlebars, glove box lid and the tail portion. The sequence is then : dual seat ; silencer ; lifting handle ; instrument panel ; front shield ; handlebar cover ; headlamp and cowl ; complete wiring, and finally the legshields and removable side panels.

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Institute of Sheet Metal Engineering

(*Continued from page* 69)

Conference should be on an international scale and offers of papers have already been made from overseas countries as well as from the U.K.

Further invitations have been and are being sent out to known experts in this field and offers of papers or suggestions in connexion with the Conference will be welcomed by the organizing sub-committee and should be addressed to the Hon. Secretary, ISME, John Adam House, Adelphi, London, W.C.2.

SHEET METAL DATA SHEET

1

Shearing Sheet Metal

Compiled by J. W. Langton, M.B.E., B.Sc.(Lond.), M.I.Mech.E.

SUCH shearing is more complex than it may seem—here we can only try to convey the basic principles.

If the sheet metal is considered as consisting of a series of fibre layers, and passed between two cutting members to be sheared, the general action of shearing is as follows. As the cutting members come together and press on the sheet metal, distortion takes place, and as the pressure is continued, cracks start from each cutting member through the material. If the shearing conditions are good then these two cracks ultimately meet and the sheet metal is sheared. Some factors affecting the shearing quality are the clearance between the faces of the cutting members, their sharpness and quality of the material, the first being perhaps the most important. If the clearance is too great, the cracks do not readily meet and the metal between them has finally virtually to be torn and a rough cut results. If the clearance is too small, then the cracks do not meet, and the metal has to be sheared through between them. Where the clearance is correct, then the cracks meet more easily and earlier. The smaller the clearance, the greater the load on the cutting machine. The amount the cutting tool has to pass into the metal before fracture takes place is called the degree of penetration, and this can vary widely with the cutting conditions and the material. Usually the thicker the metal the smaller the penetration required before fracture.

Very approximately the average degree of penetration for sheet metal is between 28 and 60 per cent.

Clearance Between Cutting Members

The best clearance for any particular circumstances can only be found by practice tests.

A rough guide is 10 per cent of the stock thickness. For harder metals nearer 6 per cent. For softer metals it would rise to 18 per cent.

Pressure for Shearing

The maximum shearing pressure is given by—

Max. shear pressure = Total length to be sheared by stock thickness by ultimate shear stress of metal.

Max. work done in shearing = Max. shear pressure by stock thickness.

or $= LT^2 f_s$ (T = stock thickness)

(L = length cut)

(f_s = shear stress).

In practice the work done would be nearer the following : Work done = $LT^2 f_s \times$ degree penetration $\times$ K, where K is a constant to allow for stripping pressure. K can vary a great deal, but approx. value would be of the order of 1.25.

SOME COMMON CUTTING METHODS

For Straight Line or Curved Cutting Portable or Hand Methods

There is a range of machines from the basic elementary snips to portable power machines. In all machines, note there are limitations to use for curved cutting and this point should be watched.

Snips can be obtained in very many sizes, and types (Universal LH—RH, etc.) up to about 16 in. long.

Scotch Shears are made up to 24 in. long, and take the form usually of a large pair of scissors.

Block Shears have one leg for insertion in vice or dolly and are made in many sizes.

Portable Ratchet Action Shears.—These machines have a geared ratchet motion worked by two long-hand levers—one for holding and guiding the machine, the other for working the ratchet feed. Made usually in two sizes—for 14 g. and $\frac{1}{8}$-in. material.

Hand Lever Shears.—Generally for bench mounting and straight line cutting. Specification in blade length, and maximum material cut. Mostly for plate rather than sheet work. Blade lengths up to 20 in. and $\frac{5}{16}$-in. plate.

Portable Power Shears.—There is quite a large range of available machines powerized by electrics or air, to cut up to about 11 g. Smallest cutting radius on lighter machines about $\frac{3}{8}$ in.

FOR STRAIGHT LINE CUTTING

Guillotines

The basic principle of these machines is that one blade is fixed, and the moving blade (inclined to the fixed blade) is brought down to meet the fixed blade.

The moving blade can be actuated by hand, foot or by power. The general specification for any of these types is the maximum length and thickness to be cut.

Hand Lever Guillotines.—These are made in various sizes and capacities up to about 40 in. in 14 gauge material, but their greatest use is for much lighter metals and much shorter lengths. Not really for cutting in any regular quantities (1)*.

Treadle Guillotines.—Where the moving blade is actuated by foot treadle. Usual maximum limitations 72 in. in 18 gauge or 48 in. in 16 gauge (2).

*Numbers refer to illustrations on the page overleaf.

1

SHEET METAL DATA SHEET

Shearing Sheet Metal (Cont.)

Power Guillotines (3).—The general powerization may be by (*a*) electric motors for mechanical guillotines, or (*b*) electro-hydraulic pumps in hydraulic guillotines, or (*c*) air cylinders from compressed air supply.

There are no real practical limits to length and capacities of (*a*) and (*b*), but the pneumatic machines are mostly confined to lengths of 72 in. by 18 gauge or 48 in. by 16 gauge. Mechanical guillotines can be obtained as overcrank or undercrank types. In the former, the crank or drive for the cutting member is above the machine bed, in the latter case below the bed. Electro-hydraulic machines are usually overcrank machines. The relative virtues of undercrank and overcrank machines are debatable, many prefer undercrank on the idea that the cutting beam guides are shorter and more rigid and also that the connections to the power unit are shorter. Since these points obviously vary with different machines, exact comparison would have to be made between specific models.

POINTS ON OPERATION OF GUILLOTINES

(1) The maximum capacity given for a machine must not be exceeded, irrespective of length, unless there is a reduction in tensile strength. Conversely, if there is any increase in tensile strength, then a larger capacity machine must be used.

Example.—For a machine having a specified capacity of 48 in. by 16 gauge mild steel.

(**Note.**—Mild steel so specified is usually of the order of 24 to 26 tons tensile.)

This machine would cut aluminium of 12 tons tensile to thickness of $\frac{1}{8}$ in., or for stainless steel (48 tons tensile) to thickness of $\frac{1}{32}$.

These remarks hold irrespective or not whether the length is 48 in. or less. Shorter lengths do not affect the point. Similarly, for cutting wire mesh, a guillotine must be selected of equivalent mild steel sheet rating as the gauge of mesh, *i.e.*, 10 gauge wire—10 gauge guillotine.

Blade clearances.—These are exceedingly important and should be set to suit the material being cut, every time. The rule given earlier (10 per cent) is approx. only, and must be varied to suit the particular material. Work with the maximum clearance to suit the quality of cut acceptable.

Guillotines will, of course, cut lighter materials than their maximum specification, provided the blade clearances are suitably adjusted.

Keep the blades sharp always.

Metal guillotines can be used for cutting non-ferrous materials, keeping the above facts in mind. For softer materials than steel, increase in the rake of the blade can usually be made with advantage.

ROTARY SHEARING MACHINES (HORIZONTAL SHAFTS)

The principle of these machines lies in a pair of cutters, one each mounted on a horizontal shaft (one shaft above the other) which overlap on the faces. The material to be cut is pushed to the overlap, and when the cutters are rotated shearing takes place (**4**).

The general specification for such machines is in the maximum thickness of material they will cut, the depth of the gap in the machine body (which controls the maximum width of one piece being sheared) and whether the shafts are hand or power driven. The machines usually have a support table for the work, which carries a side gauge working along the gap in the machine body. Fairly heavy machines can be obtained, but the most common machines work on 14-10 gauge.

The amount of overlap in the two cutters is important and this should be just slightly above the maximum thickness to be cut. In machines where there is no adjustment between the spindles, the cutters should be ground on the face only. For working on most of the light gauges, the cutters should have a face clearance of about 5 per cent to 10 per cent of metal thickness, depending on the quality of cut required.

The straightness of the cutting line depends to a large degree on the force holding the material to the gauge, but generally the cut is more curved than that say from a guillotine, but quite good enough for most purposes.

Gang Slitters

These are multi-cut rotary shears, where a number of cutters are mounted on each shaft. In the lighter machines the cutters are fixed to hubs which slide on the shafts, and in the heavier machines the cutters are screwed together with spacers between.

The machines are usually arranged to cut either sheets, or coiled material. In latter case, coil holder and re-wind coils are necessary driven from the machine itself (**5**).

Capacity specification—the number of cuts in a specified gauge of material.

MACHINES FOR CURVED LINE CUTTING

These machines will, of course, cut on straight lines also if required.

Throatless Shears.—Can be hand lever operated, or rotary type which can be hand or power operated.

For the rotary types, very often a circle cutting attachment can be obtained.

Power Rotary Shears (one inclined shaft).—These will cut on straight or curved lines (**6**). Specification as for rotary shears, *i.e.*, gap and metal thickness. Common machines up to 14 gauge. Can usually be fitted with circle cutting attachment.

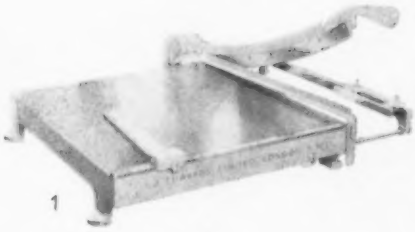
Circle Cutting Machines.—Virtually special single purpose power rotary shears as immediately above for cutting circles only (**7**). Specification—max. and min. diameter of circles and metal thickness. Hand or power operation.

Nibbler Shears.—The cutting is done by reciprocating punches, or very narrow shear blades. Machines for nibbling only are becoming obsolete, and machines are now available which will not only nibble and shear, but with additional tools perform most sheet-metal-working operations (**8**).

Presswork.—Shearing can obviously be carried out in press tools.

SHEET METAL DATA SHEET 1

Shearing Sheet Metal (Cont.)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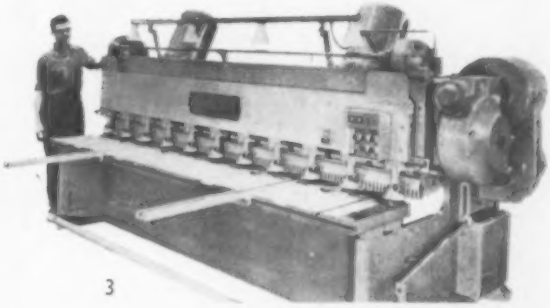
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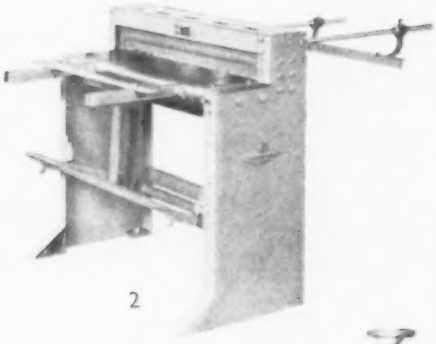
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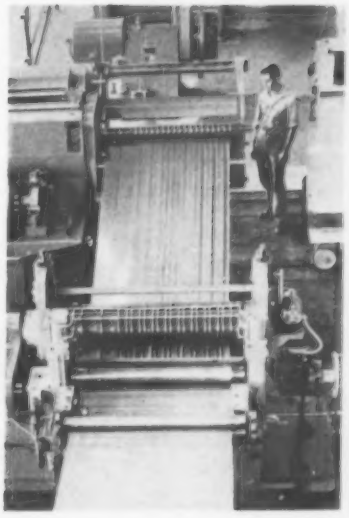
Typical machines described on two preceding pages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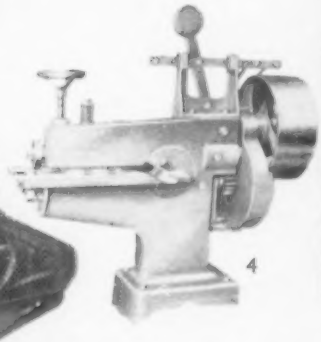
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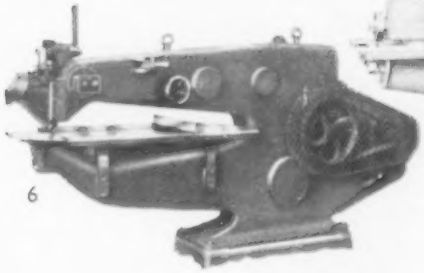
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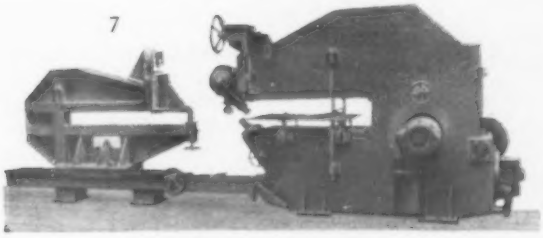
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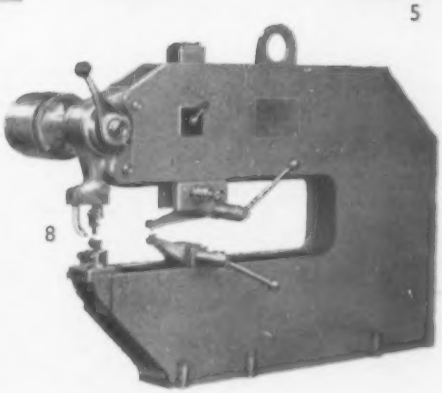
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DRUM-TYPE PICKLING OF COILED STRIP

Some Details of a French Installation

By L. W. PASSMORE

FOR plants making cold-rolled strip the problem of pickling is sometimes difficult to solve. In some instances the use of high-capacity wide strip continuous pickling lines may be uneconomical. The same applies in instances where strip of widely varying width has to be pickled, or where production is irregular.

Recently a drum-type pickler has been developed by a German firm for efficient batch pickling. The equipment is based on prolonged research carried out by Ruthner and is marketed under the trade mark " Schloemann-Ruthner " Drum Pickling Equipment.

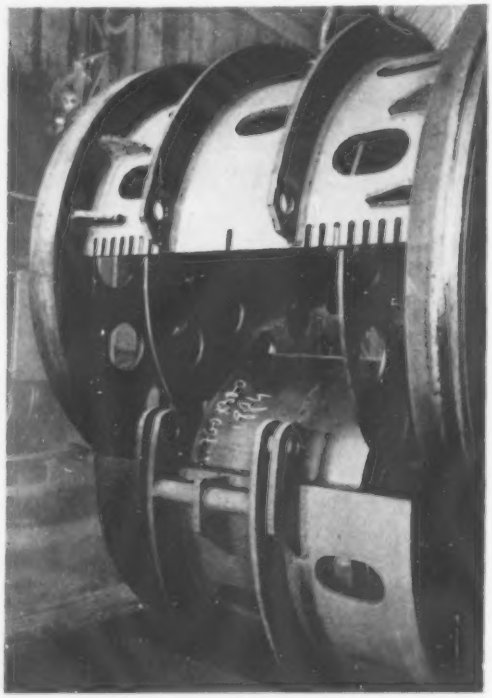


Fig. 1a (above).—Open pickling drum during charging. Cover with coil is shown at bottom

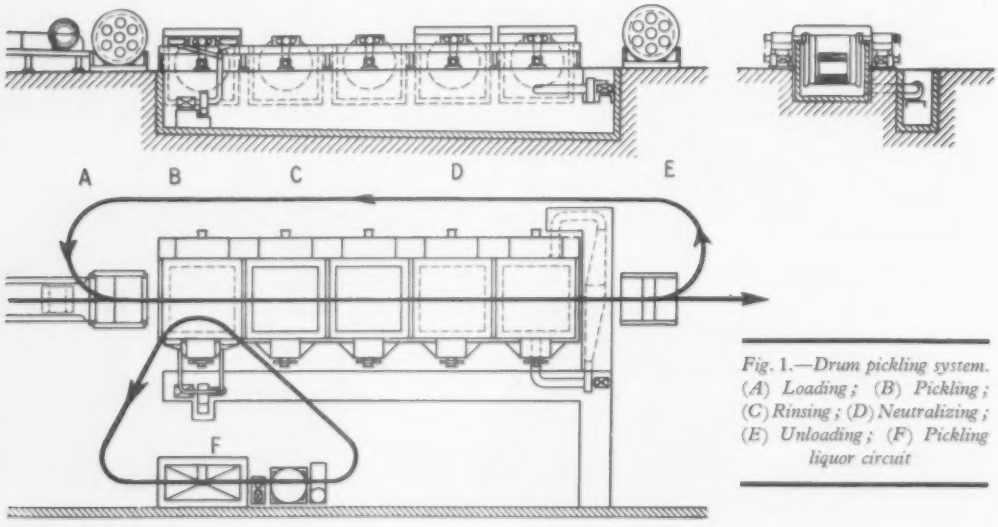


Fig. 1.—Drum pickling system. (A) Loading; (B) Pickling; (C) Rinsing; (D) Neutralizing; (E) Unloading; (F) Pickling liquor circuit

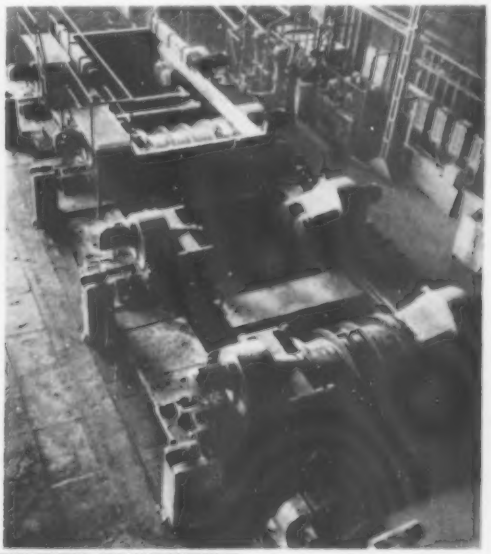


Fig. 2 (top, left).—Open drum with uncoiled coil windings

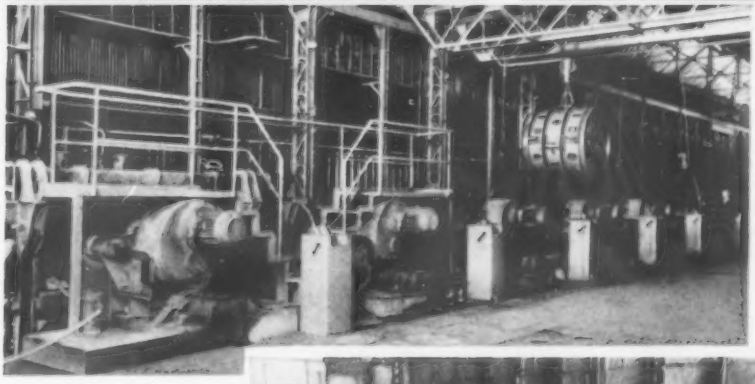


Fig. 3 (top, right).—Part view of drum pickling plant

Fig. 4 (above).—Operating side of plant showing drum with coil being shifted

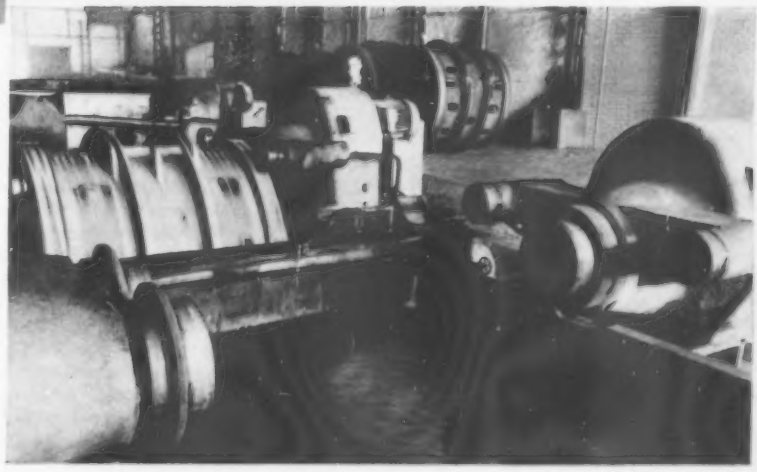


Fig. 5 (right).—Pre-rinsing vat with gear drives

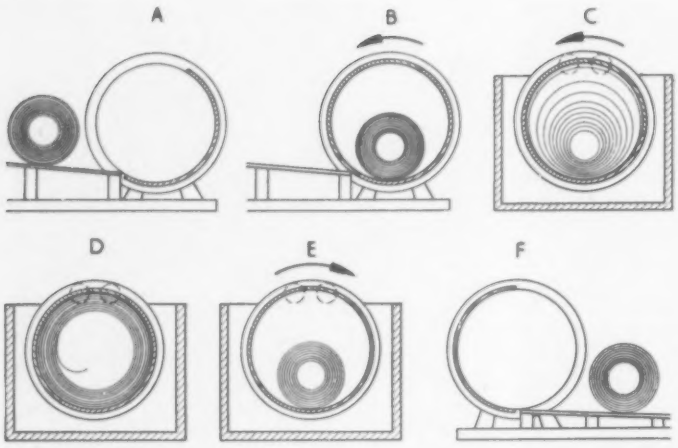


Fig. 6 (left).—Working principles of the drum-type pickling system

Fig. 7 (below).—Pickling gear drive and rotating drum in neutralizing vat

Fig. 1 illustrates schematically a complete drum pickling system. It consists mainly of vats for pickling, rinsing, neutralizing and oil emulsionizing, some drums of special design allowing uncoiling, and the necessary drive mechanism. The coil to be pickled—or several coils in case of small strip—is placed in a drum (Fig. 1*a*) and then rotated in the vats. During this process, the coil windings open to allow unhindered access of the solutions to the strip surface. After all vats have been passed, the strip surface is well cleaned. The biggest plant of this type so far built was for a rolling mill in France (*see* Figs. 2 to 5). Coils having an average width of 41 in. and a weight of 3 to 6 tons are being pickled here. The monthly throughput at the moment is 3,000 to 4,000 tons in one-shift operation; these figures can be increased.

Drum-type pickling machines for strip from 2 in. up to 79 in. in width require little space and can be easily enlarged. The pickled strip is completely free of scale and without spots. The capital investment required is low and only a few operators are needed, resulting in a highly economical installation.

Fig. 6 illustrates the flow of a typical pickling plant. The coil is inserted into the drum (A). The latter revolves first in one direction until the outer coil diameter equals the drum diameter. Then direction of rotation is reversed (B and C). This causes contraction of the coil down to nearly the original diameter (D and E). This is repeated several times and provides intimate contact of the pickling liquor with the entire coil surface. As an average the process lasts 15 to 20 minutes for de-scaling.

The drum with the coil is then transferred to vats for rinsing, neutralization, oiling and/or drying. In all these the unwinding and re-winding of the coil is performed as described before for pickling. Then the drum is emptied (F) and the treated coil is ready for further processing (Fig. 7).

The pickling liquor is cleaned, filtered and circulates in continuous flow from the pickling vat *via* a sediment vessel, re-heater and re-generator back to the pickling vat. When necessary acid is added from a measuring vessel above.

The makers claim a perfect finish of the strip, achieved within a small space, as compared with continuous plants. Acknowledgement is due to Schloemann A.G., Düsseldorf for data and illustrations.

The plant described was originally in use by Fabrique de Fer de Maubeuge, Louvroil (Nord). Later on the equipment was bought by another firm, because the plant at Louvroil was closed down. The present users, who are very satisfied with the performance, are the Société Anonyme des Fonderies, Laminoirs et Ateliers in Biache St. Vaast/Frankreich.

INSTITUTE OF SHEET METAL ENGINEERING • *Forthcoming Activities*

"PRESSES AND PRESS SHOP EQUIPMENT"

Midland Branch Discussion Forum

The next meeting of the Midland Branch will be held as usual in the Byng Kenrick Suite of the Birmingham College of Technology, Gosta Green, on Wednesday, January 13, 1960, at 6.45 p.m. Light refreshments will be available prior to the meeting. A particular welcome is extended to visitors.

The members of the expert panel who will be available to discuss questions put to them from the floor, and the subjects with which they will deal are as follows :—

Presses	Mr. A. Valance (Hordern, Mason and Edwards Ltd.).
Press Brakes and Shearing Equipment	Mr. Crosthwaite (Bronx Engineering Co. Ltd.).
Roller and Flex Levelling	Mr. F. A. Battey (Head Wrightson Machine Co. Ltd.).
Feeding Equipment	Mr. F. Evans (Press Equipment Ltd.).
Safety Equipment ..	Mr. K. H. Moore (J.P. Udal Ltd.).

Anyone interested in attending this meeting is invited to communicate with the Hon. Secretary of the Midland Branch of the Institute, Mr. E. N. Salmon, The Austin Motor Co. Ltd., Longbridge Works, Birmingham, 31.

INSTITUTE LUNCHEON

The next in the series of informal luncheons organized by the Institute will be held at the Cafe Royal, Regent Street, London, W.1, on Wednesday, January 27th, 1960, at 12.30 for 1 p.m. On this occasion the principal guest and speaker will be E. H. C. Leather, Esq., M.P., who will speak on the economics of industrial expansion. Mr. Leather is well-known for his forthright comments on many aspects of the workings of industry, and he will undoubtedly have much to say of interest to members and their guests.

These luncheon meetings have provided opportunities in the past for members to meet informally and discuss matters of common interest and the Institute has been privileged to entertain a number of eminent personalities on such occasions.

Notices of this Luncheon have been circulated to all members. Others interested in attending are invited to apply for tickets (price 27s. 6d., exclusive of drinks) to the Hon. Secretary, I.S.M.E., John Adam House, Adelphi, London, W.C.2.

COLLOQUIUM ON FORMING AND TESTING OF SHEET

As previously announced a colloquium on sheet metal forming with special emphasis on methods of testing organized jointly by the Société Française de Métallurgie and the International Deep Drawing Research Group will be held in Paris from the 23 to 25 May, 1960, inclusive. The first two days of this meeting will be devoted to open sessions at which papers will be presented and discussed on Theoretical Studies of Sheet Metal Forming, Methods of Sheet Metal Testing, and New Techniques of Sheet Metal Forming. The final day will be taken up with closed meetings between delegates of member bodies of the I.D.D.R.G. at which the programme of investigational work for the ensuing year will be drafted.

A number of papers has been offered by members of the British Deep Drawing Research Group of the Institute of Sheet Metal Engineering, which is the member organization for the U.K. of the International Group, and details of the meeting will be distributed to all members of the Group in due course to provide them with the opportunity of attending the open sessions.

Requests for further information can be made to the Hon. Secretary of the Institute of Sheet Metal Engineering or to either member of the joint organizing Secretariat as follows :—

S.F.M. : M. M. Dupont, sécrétaire-général de la Société Français de Métallurgie, 25, rue de Clichy, Paris, 9ème.

IDDRG. : Dr. S. Garber, BISRA Laboratories, Hoyle Street, Sheffield, 3, Yorks. (England).

INTERNATIONAL CONFERENCE ON COLD EXTRUSION OF STEEL

Following the Conference organized by the Institute on The Cold Extrusion of Steel in 1953 interest in the process has been slowly growing in this country. There have been certain new developments in the field of tool materials, press equipment and applications, and plans are advancing for the staging of a further conference on the subject in the late Autumn of this year. It is proposed that this

(*Continued in page* 62)

SHEET METAL NEWS

FEATURING EVENTS AND PERSONALITIES IN THE INDUSTRY

ALUMINIUM UNION NOW ALCAN (U.K.) LTD.

IT was announced recently that Aluminium Union Ltd. has changed its name to Alcan (U.K.) Ltd.

Aluminium Union Ltd. has for many years been the United Kingdom distributing company for the aluminium ingot and magnesium, bauxite and chemicals produced by its associated companies in Aluminium Ltd. of Canada.

The ingot metal bears the trade name "Alcan" and the Canadian producer has also come to be known by that name throughout the world. The new name will thus serve to give the United Kingdom company an immediate identification with the brand of metal supplied, and with the Canadian producer.

Do metal finishers want City and Guilds exams?

Borough Polytechnic principal poses question

IF any industry is to survive, let alone go forward with fresh plans for the future, it must have access to a stream of practically trained and theoretically qualified young men. With this in mind, the Institute of Metal Finishing and the City and Guilds of London Institute inaugurated two courses to provide training and subsequent examination in metal finishing practice and technology.

From the point of view of candidates taking these examinations, the response from industry has been disappointing. The figures for the last three years show a decline from 40 candidates to 25 taking the certificate in "practice," while during 1959 only three candidates have sat for the examination in "technology."

Small wonder that Dr. James E. Garside, principal of the Borough Polytechnic in London, asks whether these two courses are wanted by the industry.

Referring to the possibility that the City and Guilds of London Institute may have to discontinue holding the examinations because of lack of support, Dr. Garside thinks that if this happens it will be a "national calamity."

He suggests that all industrial executives in the metal finishing industry should send some of their employees to selected technical college courses in the subject, before it is too late.

Although the C. & G. Institute does not insist that a candidate for these two examinations should have attended a college course in the subject, Dr. Garside points out that a good deal of preparation and considerable expense have been incurred by the colleges in providing these courses. Lack of support from the industry now could well have a disastrous effect in the future.

H. & H. PRESENTATION DINNER

FAMOUS as makers of resistance welding and heating machines, Holden and Hunt Ltd. recently gave a second dinner and dance to employees at the Civic Hall, Old Hill. Watches were presented for long service to 15 employees by joint managing directors, Mr. Holden and Mr. Williams.

WEST GERMANS VISIT BROCKHOUSE

COMMERCIAL counsellor of the West German embassy in London, Dr. H. J. Mangold, accompanied by Dr. Stappert, head of the commercial department of the embassy, and a number of German journalists were guests of the Brockhouse Organization at the Group's head offices and works at Hill Top, West Bromwich, recently.

After lunch with the directors, the visitors toured the various divisions of J. Brockhouse and Co. Ltd., and were later shown over the vehicle spring making factory of Richard Berry and Son, one of the organization's subsidiary companies which has one of the most modern plants of its type in Europe.

H. PELS TAKE OVER MACHINE TOOL AGENCIES

THE machine tool agencies formerly held by K. S. Paul (Machine Tools) Ltd. have now been taken over by Henry Pels and Co. Ltd., 32-38, Osnaburgh Street, London, N.W.1.

K.9 Steel Press Tools For British Railways*

THE illustrations in the adjoining column show press tools manufactured in Edgar Allen K.9 oil hardening tool steel, for the British Transport Commission, Western Region, Reading, Berks. The tools designed by the signal engineer's jig and tool design draughtsmen were manufactured in the Reading signal works.

Combination tool

Fig. 1 shows a combination press tool for 7 in. semaphore lamp tank top and tank bottom. The tank top is blanked, raised, cropped and pierced at one stroke of the press, the material being 22 s.w.g. (.028 in.) tinplate. As regards the tank bottom, the piercing punches are easily removed in one unit so that the tank bottom may be blanked, raised and cropped at one stroke of the press.

The finished diameter of the components is $6\frac{5}{8}$ in. and 40,000 pairs are pressed between regrinds.

Piercing and raising

Fig. 2 shows a piercing and raising tool for a 7 in. semaphore lamp back frame. This tool pierces a 4 in. dia. hole and raises three sides of the blank. The material is 26 s.w.g. (.018 in.) tinplate. The size of the component is $4\frac{1}{2}$ in. by $4\frac{1}{8}$ in., and 100,000 off are obtained between regrinds.

Piercing and notching

Fig. 3 shows a piercing and notching tool for a 7 in. semaphore lamp interior body. The tool cuts 11 notches, one hole of 4 in. dia. and two slots in the blank at one stroke of the press. The die is made in three sections with one small insert. The punches are set in "Cerromatrix" alloy, with the exception of the 4 in. dia. punch.

The material is 26 s.w.g. (.018 in.) tinplate). The size of the blank is $13\frac{3}{4}$ in. by $5\frac{1}{16}$ in., and 100,000 are obtained between regrinds.

Piercing and blanking

Lastly is shown the piercing and blanking tool, Fig. 4, for producing coil cheeks for a block telegraph instrument. This is fitted with trigger stop for continuous feed. The material is $\frac{1}{16}$ in. "Dellite." The pierced holes have a diameter of .089 in. and 70,000 pieces are obtained between regrinds.

* Reprinted by kind permission from Edgar Allen News, October, 1959.

Pressed Steel Factory, Swansea

WHEN in March this year, Mr. John Rodgers, Parliamentary Secretary to the Board of Trade, laid the cornerstone of the new Swansea factory to be leased by Wales and Monmouthshire Estates Ltd. to the Prestcold Division of the Pressed Steel Co. Ltd., it was announced that limited production was due to begin in the new works in March, 1960.

With the halfway mark reached, structural steelwork for the main production building is now all but complete—itself a major achievement by the company responsible for its fabrication and erection, T. C. Jones and Co. Ltd., members of the 600 Group.

The steelwork has been welded throughout, thought to be the first time that welding has been applied to a structure of this type and size.

Vacuum Brazing in Production

U.S. aircraft company using process

IN the October issue of this journal (p. 698) a new vacuum brazing method for forming honeycomb structures was described. We now hear that the Grumman Aircraft Engineering Corpn., Bethpage, L.I., New York, U.S.A., has just ordered from the F. J. Stokes Corpn., Philadelphia, equipment for making the material on a production basis.

The equipment will be capable of brazing panels in excess of 50 sq. ft. in area. It will be used, according to Grumman, for fabricating structures for the "Eagle" air-to-air missile, a recently disclosed project being developed by the U.S. Navy's Bureau of Aeronautics.

Details of the Eagle's design and performance are still secret. Bendix Aviation Corpn. is the prime contractor for the "Eagle" project, with Bendix Systems Division, Ann Arbor, Michigan, as system manager. Grumman holds a major sub-contract from Bendix for the missile's airframe, propulsion system and ground-handling equipment.

Grumman are thus the first firm in the aircraft and missile manufacturing industry to utilize on a production-scale the Stokes-Bukata vacuum brazing process for honeycomb structures.

The process utilizes a new procedure that avoids the need for the conventional welded "envelope" to enclose the work. This permits the panels to be rapidly loaded and unloaded. The high-vacuum atmosphere created in the Stokes equipment also eliminates the need for inert gas. Both the core and the skin in stainless steel honeycomb structures are very prone to form oxides that resist the achievement of a good bond. The use of inert gas such as argon merely substitutes a non-reactive molecule for an air molecule, but the vacuum method, on

(*Continued in following page*)

Prefab. Hanger

THE American Antarctic task force will transport a prefabricated hangar to Scott Base for this year's RNZAF Antarctic flight. The New Zealand Ministry of Works has designed the exterior sheathing of the hangar in 3-ply with aluminium coating, and the frame is being built in Australia by the firm which constructed a similar hangar for the Australian Antarctic base at Mawson.

SANDERSONS EXPAND GRINDING FACILITIES

PRODUCTION facilities in the machine knife department have been expanded recently at Sanderson Brothers and Newbould Ltd. with the addition of six precision grinding machines. These are:—

A large 32 in. head segmental grinder for flat siding, edging and bevelling shear blades, guillotine knives and similar blades.

Vertical head segmental grinder for sharpening and bevelling shear blades, guillotine knives and straight knives in general; vertical head segmental grinder.

Main use will be edging and flat siding ground gauge steel before finishing this material on open-side surface grinders, but some knives will also be ground on this machine.

An open-side surface grinder fitted with specially designed magnetic chucks, capacity up to 18 ft. long and up to 18 in. wide.

A heavy-duty universal grinder for the production of slitting cutters and other circular plates up to 24 in. dia.; and an open-side surface grinder to produce ground plates up to six ft. by three ft. (*seen in picture above*).

MUREX TAKE OVER DONALD ROSS

THE whole of the issued shares of Donald Ross and Partners Ltd., Crawley, Sussex, specialists in the design and manufacture of mechanical aids to production and in particular the well-known "Twinner" positioners for hand and automatic welding, have been acquired by Murex Welding Processes Ltd., Waltham Cross, Herts.

Donald Ross and Partners Ltd. will continue to operate as in the past and under the same name and the same management.

Vacuum Brazing in Production

(*Continued from preceding page*)

the other hand, is a truly inert atmosphere.

The pilot-scale prototype equipment (shown in the photographs below) has already been thoroughly proven and is currently in regular operation in the Stokes laboratory.

Specimen panels (1 ft. by 2 ft.) brazed in the pilot-scale equipment have been furnished to many of America's leading aircraft manufacturers, where they are under test.

The equipment initially will operate at 1750° F., high enough for all present honeycomb work being done in the industry, and can be operated successfully at temperatures in excess of 2100° F.

The Stokes-Bukata equipment incorporates provision for cooling of the work once the brazing material has fused. This feature is especially desirable with precipitation-hardening type honeycomb materials, such as 15-7 Mo. and 17-7 PH stainless steels.

Both Stokes and the process inventor, Stephen Bukata, a consultant on honeycomb brazing, think the process will also be successful with materials such as titanium, molybdenum and beryllium, where the oxygen sensitivity is even greater.

Upper left—Overall view of the unit capable of brazing test-size panels 12 in. by 24 in. by 1 in. thick. Cylindrical chamber rolls horizontally on twin rails, opens for loading by moving to the left, then closes against a fixed end-plate for evacuation

Lower left—A cover sheet is placed over the honeycomb panel. An O-ring gasket runs around the perimeter of the water-cooled heating frame and forms a vacuum-tight seal with this cover sheet. A similar sheet and O-ring are located beneath the graphite reference base. Together, they form an "envelope" completely enclosing the panel. When evacuated to a high vacuum, this provides an inert atmosphere suitable for brazing—without a separate welded "bag"

Upper right—Close-up of the loading operation. Stephen Bukata, inventor of the Stokes process, is placing a tacked-together honeycomb "sandwich" upon the reference base, a block of graphite, which may be either flat or contoured as required. Heating elements extend across the lower heating tray, all along its length. Duplicate array of heating elements is provided in the upper heating frame which is mounted within the chamber

Lower right—With the cover sheet in place, the chamber is closed against the fixed end-plate and evacuation can begin. During the brazing process, hydraulic cylinders in the top of the chamber force the upper heating frame down on to the cover sheet, firmly compressing the O-ring gaskets and effectively sealing the "envelope" around the honeycomb panel

New Lease of Life

ABBEY MILL FOR NEW CANADIAN STEELWORKS

THE old slabbing mill, recently replaced by a new universal mill at the Abbey Works of The Steel Co. of Wales, will soon be back in production again.

It has been purchased by Frank B. Foster International Inc., of Pittsburgh, U.S.A., who are dealers in industrial plant and rolling mills, and was loaded on the s.s. Montreal City at Swansea for shipment to Duluth, Minnesota. Its eventual destination is Canada, where it will become part of a new works being erected by the Inter Provincial Steel Corporation at Regina, Saskatchewan. The works will produce steel for a pipe-line to carry natural gas to Canada's principal cities and industrial areas.

During its eight years at Abbey Works, the mill originally supplied from America rolled 12½ million tons of ingots into slabs. Its dismantling, and the erection of the new mill in 12½ days last November, was a considerable engineering feat.

Packing of the mill and spare parts (a total weight of nearly 1,000 tons) for the long journey, was a task undertaken at the Abbey site by the St. Tydfils Packaging Ltd., from Dowlais, South Wales.

The total packaging took about 12 weeks, and required 60,000 feet of timber and 500 gallons of paint and rust preventive. About 200 items were involved, needing 12,000 man-hours of labour.

For the final journey in this country, to the dockside, about 60 rail trucks were required, with the major items going by road. These included two mill housings weighing 112 tons apiece ; two sole plates, each of 60 tons ; a manipulator head of 56 tons and one screwdown platform (12 tons). The deck of the s.s. Montreal City was specially strengthened to carry the great weight of the two mill housings.

R.C.I. TO DISTRIBUTE PHILIPS' PRODUCTS

THE sole distributors in the United Kingdom for the industrial products at present marketed by its associated company, Philips Electrical Ltd., are now Research and Control Instruments Ltd., Instrument House, 207, King's Cross Road, London, W.C.1. A new division has been set up by R.C.I. for this purpose.

Since the specialized staff from Philips will conduct the business of the new division, there will be no break in the continuity, and customers' orders will be dealt with by the same individuals as hitherto. Fully supported by the manufacturers, R.C.I. will also take over the responsibility for the maintenance of existing Philips industrial plant.

FOUR MILLION ORDER FOR DAVY UNITED

THE order for the building of the 68-in. semi-continuous hot-strip mill for the Ravenscraig Works of Colvilles Ltd. has been placed with Davy and United Engineering Co. Ltd., Sheffield. The value of the contract exceeds £4 million.

Threefold Increase

When the new mill goes into production it will have an initial output of 500,000 tons of strip steel products a year. It is, however, being designed to provide for an eventual output in excess of one-and-a-half million tons a year. More than 47,000 h.p. will be needed to drive the mill, which will produce strip and light plate from 24 in. to 60 in. wide, in coils weighing up to 40,000 lb.

Glasgow Works Order

The contract will have an immediate impact on Scottish industry, for Davy-United's Glasgow Works will undertake a substantial part of the design and manufacture of the 9,000 tons of machinery involved. Colvilles Ltd. are themselves building a 52 in. by 138 in. slabbing mill which will supply the heavy slabs required to feed the hot strip mill.

Contribution to Industry

This new mill will be a potent force in Scotland's industrial fortune. From it will come the increased supplies of strip and sheet steel vital to the development of lighter non-traditional manufacturing industries, now recognized as increasingly essential for future Scottish prosperity.

By their introduction of continuous strip rolling facilities, Colvilles are also providing a significant step towards greater diversification within the Scottish steel industry itself.

GAS CLEANING PLANT FOR SCOTLAND

A SUBSIDIARY of Head Wrightson and Co. Ltd., Head Wrightson Iron and Steel Works Engineering Ltd., have received an order from Colvilles Ltd. for the supply and construction of a blast furnace gas cleaning plant at Ravenscraig in Scotland. The plant will mostly be manufactured by Head Wrightson at their works on Teeside. The value of this contract is approximately £357,000.

NEW COMPANIES

Ltd." is understood, also "Private Co." Figures = Capital, Names = Directors, all unless otherwise indicated.

D. & H. STEEL FABRICATIONS, 8a, Station Road, S.E.25. October 6. £100. *Christopher Huffey, Basil B. Dross.*

STEEL STRIP, 12, Ranmoor Cliffe Road, Sheffield. October 6. £100. *To carry on bus. of manufacturers of and dealers in saw and spring steels, etc. Directors not named.*

WILSON AND WRIGHT, Carr Lane, Glasshoughton, Castleford. October 6. £2,000. *To carry on bus. of scrap metal merchants, metal and machinery merchants etc. Jas. Lumb, Mrs. Elizabeth Lumb, Beryl Holmes, Harry Wright, Alfred Wright.*

EAST ANGLIAN METAL MERCHANTS, 66, Cottesmore Gardens, Leigh-on-Sea. October 7. £1,000. *Derek L. Kemp, Joan M. Kemp, Charles R. Turner.*

HEATED HOSES, Milton Works, Egerton Lane, Sheffield, 1. October 8. £2,000. *To carry on bus. of general engineers and gauge makers, etc. Peter M. Pathe and Barbara G. Pathe.*

METALLIC TOOLS AND MACHINERY, Kingsbourne House, 229-231, High Holborn, W.C.1. October 9. £1,000. *Ronald G. Cook, Norman A. Wrake.*

WIRE WELD (LONDON), 198-204, Lewisham High Street, S.E.13. October 9. £1,000. *To carry on bus. of iron and brass founders, metallurgists, etc. Derek J. Sales, Denis North, Kenneth J. Castle.*

MAUVLEY BROWN, 131a, Eltham High Street, S.E.9. October 12. £100. *To carry on bus. of boilermakers, welding, mechanical and general engineers, etc. Joseph M. Brown, Mrs. Blanche A. Brown.*

ROBINSON'S STEEL STOCKHOLDERS (NETHERTON), Podmore Road, Woodside, Dudley, Worcs. October 12. £3,500. *Ernest Robinson, Donald W. Robinson, Keith D. Robinson.*

WELLS AND HINGLEY, 7-8, Nelson Works, Back Road, Sidcup, Kent. October 14. £1,500. *To carry on bus. of manufacturers and repairers of moulds and machinery and dealers in plastics, etc. George A. Hingley, William G. Wells.*

DAGENHAM WIRE WORKS, 73, Cheapside, E.C.2. October 15. £1,000. *Arthur H. M. Clifford, May A. Clifford and David B. Clifford.*

From the Register compiled by Jordan & Sons Ltd., 16, Chancery Lane, London, W.C.2.

Exhibition in the U.S.A.

F.B.I. NEW YORK TRADE FAIR

IT was announced recently that British Overseas Fairs Ltd. will remain a private company but, "to provide working capital and to avoid the need for raising guarantee funds for individual projects," 100,000 ordinary shares of £1 each, 10s. paid, are being issued to certain members of the Federation of British Industries, to the F.B.I. itself, and to the National Union of Manufacturers.

B.O.F.'s main task at present is the organization of the British Exhibition and Trade Fair to be held in New York at the Coliseum Exhibition building from June 10-26 next year. To assist B.O.F. in its task, a top-level executive committee has been set up, including the following members: The Rt. Hon. Lord Rootes, G.B.E., chairman of the Dollar Exports Council, Mr. W. H. McFadzean, president of the Federation of British Industries, and Sir Norman Kipping, J.P., director-general of the Federation of British Industries and chairman of British Overseas Fairs Ltd.

B.O.F. Ltd. was originally formed by the F.B.I. to organize the British Trade Fair in Baghdad (1954). The success of that experiment resulted in a decision to retain the company for other international trade fairs such as Helsinki (1957) and Lisbon (1959).

The New York Fair will be the first comprehensive all-British exhibition and trade fair to be staged in the U.S.A., and will be held at the Coliseum Exhibition building in the heart of New York City (above) on all four floors. It will be nearly twice as large as the recent U.S.S.R. exhibition in the same building.

The U.S.A. has become our biggest single market, taking 8 per cent of our total exports.

American experts have been appointed to advise the organizers on all aspects of the Exhibition, including contractual and other problems, advertising and public relations.

The whole of the 3rd floor will be devoted to engineering. Many of Britain's most famous firms have booked space to display their latest products, and an entire section, comprising over 7,000 sq. ft., has been taken by the British motor industry, where manufacturers will show one model from each famous range of cars selling in America.

ELGAR AGENCY

THE Elgar Machine Tool Co. Ltd. have been appointed the exclusive distributors in the U.K. for the Hyop range of jig boring machines manufactured by Apparatebau Burkhardt, Pfullingen b. Reutlingen, Germany.

APPOINTMENTS and STAFF CHANGES

Left to right.—
TAYLOR CORNELIUS
R. E. VERGUSON
CECIL TRACEY

It is announced that the following members have been elected to fill vacancies on the council of **The Institute of Metals** with effect from the annual general meeting on March 29, 1960 :—

President: Sir Ronald Prain, O.B.E., Hon. M.I.M.M. (chairman and president, Rhodesian Selection Trust, Ltd.). *Vice-Presidents:* H. M. Finniston, B.Sc., Ph.D., A.R.T.C., F.I.M. (research manager, C. A. Parsons and Sons Ltd.), H. W. Hignett, B.Sc. (Eng.), F.R.I.C., F.I.M., M.I.W. (assistant managing director, Henry Wiggin and Co. Ltd.)

Ordinary Members of Council: R. W. K. Honeycombe, M.Sc., Ph.D. (professor of physical metallurgy, University of Sheffield), Ivor Jenkins, D.Sc., F.I.M. (chief metallurgist, research laboratories, The General Electric Co. Ltd.), E. Robson (managing director, The Manganese Bronze and Brass Co. Ltd.), J. Salter, B.Sc. (Tech.), A.M.I.E.E. (director The British Aluminium Co. Ltd.), Christopher Smith, F.I.M. (works director, James Booth and Co. Ltd.).

* * *

The Pressed Steel Co. Ltd. announces that Mr. Cecil F. Tracey has joined its organization as a divisional director of the Prestcold division with the title of production director of the new Swansea plant.

* * *

Dr. D. W. Davison, F.Inst.P., F.I.M., has been transferred to the central personnel executive staff at the head office of **The Plessey Co. Ltd.,** Ilford, to assist the personnel director with scientific and technical appointments throughout the company and its associates.

* * *

Mr. F. J. Slee, a well-known personality in the oil industry, has retired as joint lubricants manager of **Shell-Mex and B.P. Ltd.** after 36 years' service.

* * *

Tube Investments announces that Mr. Bryan Harris, a graduate in metallurgy from the University of Birmingham, has been appointed to a TI Research Fellowship for work in the physics and chemistry of solids, at the Cavendish Laboratory, Cambridge.

* * *

Mr. F. W. S. Hewitt, northern regional sales manager of **Domestos Ltd.,** College Works, Albion Row, Newcastle-upon-Tyne, has been appointed southern regional sales manager—industrial division of the company.

Mr. J. Harrison, who becomes Newcastle branch area manager, has been in charge of the Carlisle branch since 1953, in the dual capacity of branch area manager and industrial representative.

* * *

Northern Aluminium Co. Ltd. have accepted the resignation of Mr. C. P. Paton, who is taking up an important position with Aluminium Ltd. of Canada in Montreal.

Mr. Taylor Cornelius, who has taken over from Mr. Paton as Northern's general works manager, has been elected a director of the company. Mr. Cornelius was until recently a vice-president of Aluminum Co. of Canada Ltd. and manager of that company's fabricating division.

* * *

Mr. Tibor Haas, Dipl. Ing., A.M.I.Mech.E., A.F.R.Ae.S., has been appointed head of the design section in the newly-formed members' service department of the **British Welding Research Association,** Abington, Cambs.

Mr. Haas, who was educated at Prague University, faculty of mechanical and aeronautical engineering, has had many years' experience in mechanical engineering, light weight engineering design and heavy engineering.

Following a period as industrial consultant on contracts throughout Europe he joined the Bristol Aircraft Co. of Filton where, for the past six years, he has held the position of head of the fatigue department.

* * *

The Consolidated Pneumatic Tool Co. Ltd., 232, Dawes Road, London, S.W.6, announce the opening of a new district office in Bristol to serve industry in the west of England, under the management of Mr. J. D. Glover. It is at 1, Bedminster Down Road, Bristol, 3. Telephone : Bristol 63736.

* * *

The alloys division of **Union Carbide Ltd.** announce the following executive appointments within their sales organization :—

Mr. Colin Dunger, previously area sales manager—Sheffield, is now assistant sales manager. Mr. William Smith, who has been the company's technical sales representative in Scotland for some time, is appointed area sales manager—Scotland office: 266, Clyde Street, Glasgow, C.1.

Mr. William Paton has transferred from the division's development department to take charge of the sales department's Birmingham

Appointments and Staff Changes

(*Continued from preceding page*)

area, and is now area sales manager—Birmingham office at Beaufort House, 92, Newhall Street, Birmingham, 1.

* * *

The Plessey Co. Ltd. announce the appointment of Mr. R. E. Verguson, M.S.M.A., as commercial manager of a new grouping which includes the sheet metal division and the Amar Tool and Gauge Co. Ltd. with the industrial hydraulics division, at Cheney Manor, Swindon.

The recently formed Association of Hydraulic Equipment Manufacturers has elected Mr. Verguson to its executive committee.

* * *

The Steel Company of Wales Ltd. has announced that following the completion of the company's third development plan, and in view of the retirement of Captain H. Leighton Davies in June, 1960, the under-mentioned appointments to the board of directors have been made: Mr. R. W. Evans; The Hon. R. H. M. Kindersley, and Mr. I. S. Scott-Maxwell.

Captain Leighton Davis is to continue as a consultant to the company following his retirement.

Mr. R. W. Evans, at present assistant general manager (operations), steel division, is appointed deputy general manager, steel division.

The Hon. R. H. M. Kindersley, joins the board as a non-executive director.

Mr. Scott-Maxwell, aged 48, is appointed general manager designate to the tinplate division to succeed Captain Leighton Davies. Mr. Shirley Gazard, assistant general manager—tinplate division, is appointed deputy general manager to the division.

* * *

Mr. F. C. Braby, M.C., D.L., M.I.Mech.E. (chairman and managing director of Fredk. Braby and Co. Ltd.) has been elected chairman of the council of **The British Non-Ferrous Metals Research Association** to succeed Dr. Maurice Cook, C.B.E. who retired from office on December 31 last.

Mr. Braby has played an active part in the affairs of the Association since 1929, and has been a member of the council since 1935. In 1950 he was appointed vice-chairman and hon. treasurer.

* * *

Mr. H. P. Forder and Mr. A. R. Hay have been appointed directors of **The United Steel Co. Ltd.**

Mr. H. P. Forder is director and general manager of the Samuel Fox and Co. Ltd. subsidiary of the company. Mr. Forder joined the company in 1930 as an apprentice. He was appointed commercial manager of Samuel Fox in 1946, and a director of the branch in 1949, becoming deputy general manager in 1955. He was appointed general manager in July, 1956.

Mr. A. R. Hay is a director of Naylor, Benzon and Co. Ltd., who represent United Steels in the Newcastle area.

Mr. H. Morley has been appointed a director of Samuel Fox. General works manager, he joined the company in 1934. He became manager of the wire department in 1948 and chief work study officer of United Steel in 1954, returning to Samuel Fox in 1957 as works manager (services). He was appointed general works manager in 1958.

Mr. P. Beynon has been appointed works manager, operational research and work study at Steel, Peech and Tozer, another branch of United Steel. Mr. Beynon was previously work study manager.

* * *

The Incandescent Group of Companies (The Incandescent Heat Co. Ltd., Metalectric Furnaces Ltd., Controlled Heat and Air Ltd. and Selas Gas and Engineering Ltd.) has opened an area office at 40, Newport Road, Cardiff (Telephone: Cardiff 37715) under the management of Mr. J. W. Payne, who has been operating for the group in south Wales for the past two years.

* * *

Mr. N. R. Crump, president of the Canadian Pacific Railway Co. has been elected a director of **The International Nickel Co. of Canada Ltd.**

Besides being a director of Canadian Pacific, Mr. Crump's directorships include Bank of Montreal, The Consolidated Mining and Smelting Company of Canada Ltd. and Mutual Life Assurance Company of Canada. He is a Governor of McGill University and Royal Victoria Hospital, Montreal and a Trustee, Corporation of Bishop's University, Sherbroke, Quebec.

Steam up in Birmingham

DURING the first Saturday and Sunday in January, the City Museum and Art Gallery in Newhall Street, Birmingham 3, is holding a "steam weekend." Four of the exhibits that will be shown working are a Massey steam hammer (1873), a Galloway Uniflow engine (1924), an Amos compound beam engine (1864), and a Tangye Colonial engine (1890).

Three hot air engines and a gas engine will also be run. For short periods on each day, an Orchestrian organ and a Pollitt and Wigzell horizontal compound engine with Corliss valve gear will also be exhibited under power.

OBITUARY

Sleath: We regret to announce the death of Dennis Bernard Godwin Sleath, at the age of 45.

At the time of his death he was manager of the Birmingham area office of Borax Consolidated Ltd., a firm he joined in 1936 and with whom he had worked continuously except for a period of war service. During the war, he was commissioned in the Army Intelligence Corps.

He was honorary secretary of the midland section of the Institute of Vitreous Enamellers, and his wide knowledge of languages had stood him in good stead at the many international conferences of the Institute he had attended.

He leaves a wife and five children.

Smith: The death occurred on November 10 at the age of 51 of Mr. Frank Smith, works manager of Samuel Gregson and Sons Ltd., sheet metal workers, Bury (Lancs.). He had been with the firm for 36 years and was chairman of the Bolton branch of the Engineering Industries Association.

Seferian: Many readers of this Journal will learn with regret of the sudden death on September 29 of Daniel Seferian at the age of 59. The author of a number of standard works in the French language on welding, he was known to many in the welding industry of the U.K. for the English version of his book "Les Soudures" which appeared as a series of articles extending over many months in this Journal. Monsieur Seferian joined the French Institute of Welding in 1927 and worked on welding research under Professor Portevin for 18 years. He was awarded the degree of Dr. es Sc. in 1935 for his thesis on "Study of the Formation of Nitrides of Iron by Fusion and of the Iron Nitrogen System."

He left the Institute in 1945 to go into industry and after two other appointments he joined the Soc. Sarazin et Cie in 1952 where his scientific knowledge and long practical experience stood him in good stead as the head of the electrode research department.

Forthcoming Events . . .

January 5

Institution of Plant Engineers (Peterborough Branch). "The Use of Plastics in Engineering," by C. L. Child (I.C.I. Ltd.), at the White Lion Hotel, Church Street, Peterborough. 7.30 p.m.

January 6

Swansea and District Metallurgical Society. "Steel for Atomic Energy," by Mr. Brychan Watkins, at the Central Library, Swansea. 6.45 p.m.

January 7

Newport and District Metallurgical Society. "Automatic Control in Rolling Mills," by Peter Briggs (Davy United Engineering Co. Ltd.), at the Whitehead Institute, Cardiff Road, Newport, Mon. 7 p.m.

January 11

Institution of Mechanical Engineers (Lubrication Group). "Recognition of Deterioration of Lubricants in Service," discussion, at 1, Birdcage Walk, London, S.W.1. 6 p.m.

January 13

Institute of Welding (South London Branch). "High Temperature Brazing—A Modern Joining Method," by E. R. Perry, at 54, Princes Gate, S.W.7. 7.30 p.m.

North Wales Metallurgical Society. "Analytical Control by Means of the Quantometer and Quantovac," by W. S. Sykes (Steel, Peech and Tozer, Rotherham), in the Lecture Theatre, Flintshire Technical College, Connah's Quay. 7 p.m.

Institute of Sheet Metal Engineering (Midland Branch). "Presses and Press Shop Equipment (Forum)," at the College of Technology, Gosta Green, Birmingham. 6.45 p.m.

January 20

Institute of Welding (North London Branch). "Further Developments in the Welding of Atomic Plant," by A. Prince, at 54, Princes Gate, S.W.7. 7.30 p.m.

January 21

Institute of Metals (Birmingham Local Section). "Physical Methods of Analysis," by K. M. Bills, at the College of Technology, Gosta Green, Birmingham. 6.30 p.m.

Lightweight Welding Gun

WELDING TRANSFORMER USES ALUMINIUM WINDINGS

THE benevolent burglar seen in our picture above has just finished cutting open a safe full of Christmas toys for the children of Great Ormond Street Hospital, London.

In real life the managing director of Ashton-Young Engineering Development Co. Ltd., he was demonstrating the versatility of the company's Balarc "175" air cooled welding transformer.

Scheduled for production by the Bermondsey, London, factory of Remploy Ltd., the national employment organization for disabled persons, this 175-amp transformer uses anodized aluminium foil in its windings, resulting in a large weight saving over the conventional copper-wound unit. Its capacity is from 16 to 8 gauge inclusive, with the ability to handle 6 gauge on intermittent working. Total weight is only 67 lb., and the unit meets the standards specifications of many countries, including the U.K., U.S.A., and Germany. A two-pole, six-way roller switch controls amperage at both 80 and 50V O/C at 23V welding, running from 190 to 250V A.C. single-phase 50 cycles, and draws a maximum of 40 amps from the mains.

January 26

Sheffield Metallurgical Association. "The Analytical Chemistry of the Platinum Metals," by E. Jackson (Sheffield College of Technology), at the BISRA Laboratories, Hoyle Street, Sheffield, 3. 7 p.m.

Institution of Plant Engineers (South Wales Branch). "Safety in Uses of Industrial Gases and Equipment," by M. T. Reeks (British Oxygen Gases Ltd.), at the South Wales Engineers Institute, Park Place, Cardiff. 7.30 p.m.

January 27

Institute of Sheet Metal Engineering (Wolverhampton Section). "High Speed Blanking and Piercing and the Production of Laminations," by N. V. Larking (J. Sankey and Sons Ltd.), at the College of Technology, Wulfruna Street, Wolverhampton. 6.45 p.m.

January 28

Institution of Plant Engineers (Sheffield and District Branch). "Instrumentation," by Dr. R. H. Baulk (Samuel Fox and Co. Ltd.), at the Grand Hotel, Sheffield. 7.30 p.m.

Society of Instrument Technology (Grangemouth Section). "Computer Control of a Continuous Process Plant," by a member of Elliott Bros. Ltd., in the Ellwyn Restaurant, Newlands Road, Grangemouth. 7 p.m.

New Plant and Equipment

Cobalt-platinum Alloy Possesses Good Magnetic Properties

A COBALT-PLATINUM magnetic alloy, Platinax II, is now being produced by Johnson, Matthey and Co. Ltd., 73-83, Hatton Garden, London, E.C.1.

The outstanding magnetic properties of certain alloys in the cobalt-platinum system have been known to metallurgists for many years, but it is only in recent times that the properties of the alloy containing an equal number of atoms of each element (23.3 per cent by weight of cobalt) have been fully appreciated. Platinax II is the latest development in this field and is one of the most powerful permanent magnet materials.

After heat treatment by a special JMC process, Platinax II is said by the makers to have remarkable physical and magnetic properties, a remanence of about 6,400 gauss, a coercive force of 4,800 oersteds and a BH (max) of 9.2 by 10^6 gauss-oersteds being obtainable. Moreover, in the heat treated state the alloy is magnetically isotropic, exhibiting similar magnetic properties in all planes.

Apart from its exceptional magnetic properties, the use of Plantinax II offers a further advantage in that the alloy is comparatively workable before heat treatment and can be machined, rolled and drawn without difficulty, and so lends itself to the manufacture of magnets of complicated shape or extremely small size.

Flash Butt Welder for Flat Stock

A NEW design of flash butt welder, often referred to as a skelp welder, is being made by Sciaky Bros., Inc., 4915 West 67th Street, Chicago 38, Illinois, U.S.A.

Known as the BPW.3-400H, the machine is intended for the continuous end-to-end fastening of flat wide strip stock. In a matter of seconds, a weld can be made as strong as the parent metal, the makers say. The machine also shears off flash particles so that a continuous flush surface is maintained. Capacity of the machine is low carbon steel ranging from 0.060 in. to 0.200 in. in thickness and from 12 in. to 22 in. in width. Or, a maximum capacity of 4.5 sq. in. cross sectional area of mild steel, and 2.6 sq. in. cross sectional area of stainless steel.

The upset force is adjusted to a maximum of 90,000 lb., while clamping force is adjustable to a maximum of 135,000 lb. Maximum platen opening is 8 in., and flashing action is accomplished by the right-hand platen. Pressure lubrication can be provided.

The secondary or welding current is obtained from air cooled welding transformers, rated at 400 kVA. at 50 per cent duty cycle. Silicone insulation of adequate section is used to minimize heat loss. This design thereby removes the possibility of damage due to freezing of cooling water or failure of water supply.

Heat control, as well as post heat, are provided. Transformer tap switches offer 32 steps of heat regulation. Secondary current is provided to both ends of the lower die to ensure uniform heating in the weld area.

Sequence of Operation

Facing the machine, the strip movement is left to right. The outgoing strip is located against the positioning or gauging bar in the initial die spacing, and the left-hand platen positioning cylinders locate the edge of the strip. The left-hand clamp is then initiated. The incoming strip is located against the gauging bar and the edges aligned similarly to that of the left-hand platen. The incoming strip is then clamped.

After clamping, the gauging bar is retracted from between the ends of the strips by a hand-push button. Another push button starts the welding operation.

The flashing cam advances the right-hand platen rapidly to close the gap between the strip ends and then starts the flashing action. During the flashing time a high-current at low voltage causes the strip ends to glow and reach a plastic state. An upsetting action then forces and fuses the ends together.

Upsetting is initiated by an adjustable limit switch actuated by the flashing cam. The limit switch operates a large orifice electro-valve that rapidly exhausts the metered air on the lower side of the flashing cylinder. When the metered air is exhausted, very fast upsetting takes place due to the preloaded air pistons.

The limit switch also initiates the upset current timer controlling the duration of the upset current flow. The change from flashing current to upsetting current is accomplished without interruption.

After upsetting and an adjustable hold time, the clamps are released and the lift rollers raise the strip above the level of the lower dies. The platen returns to its initial position, and the welded strip passes through the welder to the plough shear where the flash particles are removed. A push button initiates the gauging bar, and brings it to its position in the initial die space for the next operation.

The address of the U.K. branch of Sciaky is at Falmouth Road, Slough, Bucks.

Anti-dross Compound for Solder Baths

THE reduction and prevention of oxides capable of forming dross on the surface of solder baths is an important consideration in the rapid production of quality tin boxes. It is necessary to retain the fluidity of the solder to ensure rapid penetration into the joint and a smooth good looking fillet free from pinholes.

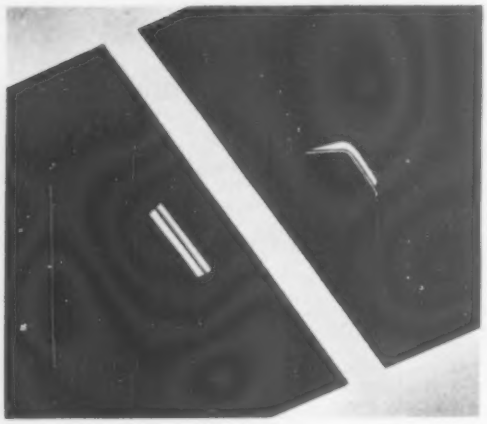
This work is generally done by the application of solid flux or anti-dross compound and is normally used on dip soldering equipment for automatic can making. Such fluxes are usually of zinc chloride base and their working life is necessarily limited in relation to the working temperature which not only governs the amount of oxide likely to be produced, but also the consistency of the anti-dross compound.

Some of these fluxes have a tendency to encourage contamination of the solder bath by reducing iron associated with the tin/iron compound upon the tinplate. Anti-dross compounds generally give rise to copious fuming which may distress operators or even attack adjacent machinery with corrosive condensates.

An anti-dross compound recently introduced by Fry's Metal Foundries Ltd., Tandem Works, Merton Abbey, London, S.W.19, is claimed to have several advantages.

Using the compound, the makers say that there is a considerable reduction in the quantity of fume evolved and an additive is also present claimed to prolong the life of the flux in a fluid condition by delaying coagulation. It also inhibits to some extent the formation of iron compound and prevents contamination which can give rise to sluggish metal and heavy unsightly fillets. The product has been given the nomenclature "A.50 compound."

(*Continued in page* 80)



Electronic Press Feed for Transformer Laminations

ILLUSTRATED in Figs. 2 and 3 is an interesting electronic feed developed by B.H.P. Machine Tool Co., 91, Wattville Road, Birmingham, 21, for use in the manufacture of transformer laminations.

The equipment is being used in a well-known electrical works in Manchester to produce laminations built up of cold-reduced, grain orientated electrical steel 0.014 in. thick, cut into lengths from 9 in. up to 12 ft., in widths between 1 in. and 10 in. The material is supplied to the works gang-slitted and coiled.

The specification to which the feed has been made is as follows: (1) Feed with constant accuracy to within 0.015 in. per length; (2) Provide for quick changeover to different lengths; (3) Give the maximum feed rate possible with this degree of accuracy; (4) Decoil and feed without marking the material; (5) Accommodate varying widths of material.

For convenience, an existing mechanical press was used to crop the ends and produce the tongue and groove shown in Fig. 1. For the feed, B.H.P. have used the motorised roller principle, controlled electronically and using magnetic clutches and brakes. The "Calpef" feed unit, as it is called, with the coil cradle, mechanical press, and conveyor belt, can be seen in Fig. 2. The required length of feed is pre-set on a Londex electronic controller, the coil cradle drive being operated by a time switch set to balance up with the Calpef unit, so that a reservoir of material is always available to avoid any drag or snatch that could affect the feed accuracy.

Adjustable coil guides are provided to suit the coil width used, and the strip is taken into power-driven feed rollers in the Calpef unit. At the outlet side, the strip passes beneath a sensitive roller on a pivot arm that lifts and operates a safety switch should the strip buckle slightly because of obstruction in the tooling. The strip passes through guides into the tool and out to a small conveyor belt that carries the lamination away to a pallet.

Drive from the motor of the feed unit is transmitted through a Warner electro-magnet clutch (Westool Ltd.) to a reduction gearbox that provides a strip feed rate of 40 ft. per min. To ensure an accurate cut-off length, a secondary gearbox is also incorporated, brought into operation by another electro-magnetic clutch to reduce the rate of feed towards the end of the movement, and finally a brake is applied. To achieve a smooth change-over to slow speed, a voltage control is used.

To set the required strip length, the Londex control panel has four switches each with a dial counting to 10; the first represents 10 units, the second 100, and so on, each unit representing 0.015 in.

With the line in operation, the decoiler, Calpef feed unit, cropping unit and disposal conveyor work in cycle automatically. If necessary, a Londex batch counter can be used to switch off the feed line when the required number of laminations have been produced.

Fig. 1 (top left).—This simple tongue and groove system was used to join the ends of material that had previously been gang-slitted and coiled

Fig. 2 (bottom left).—The Londex electronic controller at the rear of this picture controls the "Calpef" feed unit

Fig. 3 (above).—The complete line comprising decoiler, feed unit electronic controller, cropping unit and disposal conveyor

AUTOMATIC FEED PRESSES

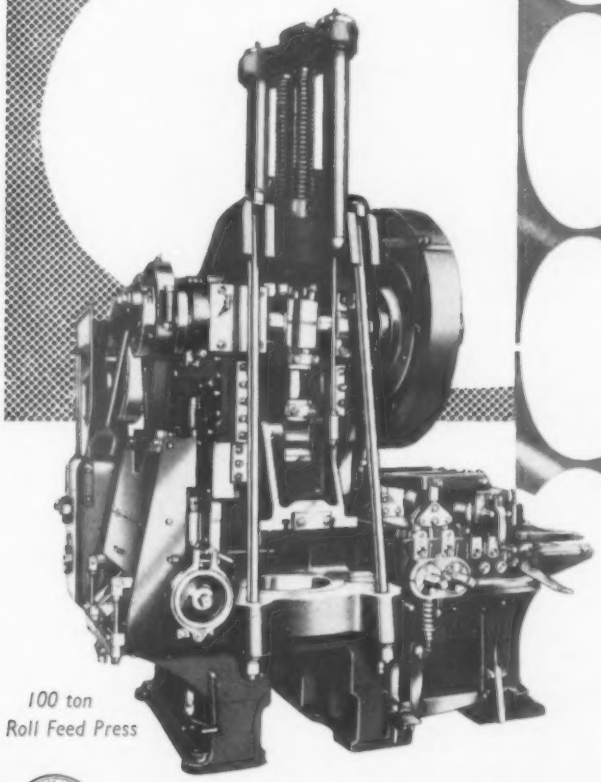


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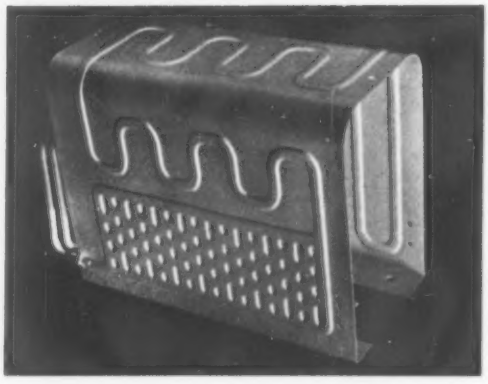
Fig. 4 (left).—This flexible ball mandrel can give bends of a radius only twice the diameter of the tubing

Fig. 5 (below).—Solid aluminium sheet having internal systems of passages and ducts



New Type Ball Mandrel

A NEW flexible ball mandrel has been developed at Boeing's Wichita plant for making bends in thin-wall tubing. Called the "Free-Flex" mandrel, it is claimed to be easy to use and more durable and versatile than any of its predecessors. Fig. 4.

The new mandrel has a threaded pin with a spherical head to interconnect a series of steel balls. The head of the pin fits into a spherical seat in the ball, permitting movement in any direction. This design and construction combines the flexibility of a cable mandrel with the strength of a link mandrel.

One of the important features of the new mandrel is that it is said to give top-quality bends of a radius only twice the diameter of the tubing.

A licence to manufacture the new mandrel has been granted to Pines Engineering Co. Inc., Aurora, Ill., U.S.A. Mandrels are available in diameters ranging from 1 in. to 6 in. for use in bending non-ferrous as well as stainless steel tubes.

Alloy Produced by Powder Metallurgy

A MACHINABLE, high density material composed of tungsten, nickel and copper and made by a powder metallurgy technique, "Mallory 1000" is now being produced and marketed by Johnson, Matthey and Co. Ltd., 73-83, Hatton Garden, London, E.C.1. It has a uniform structure, high strength and great density. As the material may be subjected to all normal machining and grinding processes, it is suitable for the manufacture of components where high mass coupled with high strength and small volume are important considerations.

Components fabricated from Mallory 1000 are finding increasing application as counterbalances for aircraft control surfaces, balances for flywheels, inertia members in such instruments as accelerometers, and as rotating inertia members, where strong precision-machined rotors are required in a limited space.

A further major application of Mallory 1000 is in radiation shielding. By virtue of its physical structure and its very high density, Mallory 1000 effectively absorbs gamma rays and can also be used for the shielding of neutron-emitting fission by-products, provided that the intensity of bombardment is relatively low. Such low-intensity neutron bombardment is said to cause only moderate activity in the material.

Mallory 1000 is supplied in special pressed shapes as well as in the form of round and rectangular bars.

Sheet with Integral Ducts

ILLUSTRATED in Fig. 5, "Noralduct" is solid aluminium sheet having internal systems of passages and ducts. It is particularly suitable for heat transfer equipment and, as the integral passages can follow almost any complex continuous pattern that can be drawn on paper, there are many other possibilities.

Sheets of Noralduct can be formed after manufacture, as are those used in refrigerator evaporator components, for example. Either a plain or an impressed pattern finish is available and colour can be included in the product design as there are no welds to spoil the appearance of an anodized finish. Makers are the Northern Aluminium Co. Ltd.

DO YOU BUY THE BEST ELECTRODES ?

Which are the best welding electrodes? In the words of the philosopher, it all depends on what you mean by best. Best for your welders? Best for the job? Or best for your company? Best from the welder's point of view are electrodes giving ease of manipulation, speed of deposition, minimum spatter and easy slag removal. Best for the job are good mechanical properties, absence of undercut and welds of neat and regular appearance. Best from your company's point of view are high quality electrodes at a competitive price backed by good service

When selecting the most suitable electrode for the job many engineers find it best to rely on their own practical experience and that is why most of them select the Murex "Fastex 5" electrode as the best general purpose electrode for the welding of mild steel.

APPLICATIONS AND ADVANTAGES

The two main features which make the "Fastex 5" type the most popular electrode with welders are its smooth running properties and ease of operation. Added to these features are the advantages of self detaching slag and the very neat appearance of the weld deposit. The electrode is therefore eminently suitable for general fabrication work, particularly where horizontal fillet welding is required. The deposited metal can be hot forged.

There is little loss due to spatter and weld cleaning operations are reduced to a minimum. As a result of the correct distribution of heat between molten slag and weld metal the control of weld contour is easily obtained when using heavy gauge electrodes so that comparatively small fillet welds can be made with consequent saving in the cost of production.

The 14 S.W.G. electrode is particularly suitable for welding light gauge sheet metal, especially for making fillet welds. Also as touch welding is relatively easy to carry out, the "Fastex 5" electrode is an ideal type for initial training in welding.

Although primarily designed for flat and horizontal vertical welding, the "Fastex 5" electrode can be used in other positions. This is why it is so useful as a general purpose electrode for so many different applications. It is also fully approved by all the main approval authorities. In short, it is the best electrode for most jobs.

A typical application of Murex "Fastex 5" electrodes. An all-welded integral body frame of a multiple unit diesel train designed and built by D. Wickham & Company Limited, Ware, Herts.

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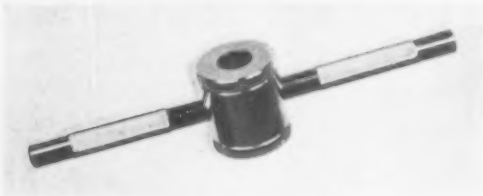


Fig. 6.—An electrode-tip dressing tool

Welding Equipment

THE basic range of portable spot welding guns produced by Portable Welders Ltd., Buckingham, has been expanded and there is now a choice of 12 heavy duty models, available both for hand, pneumatic and air/hydraulic operation. The new Autospot 4.8 kVA. machine will continue to be sold as a general maintenance and tack welder.

The company have recently put on the market two new products; one of these is a simple tip dressing tool to dress centre tips accurately without removal from the electrode arms (Fig. 6). There are two models, one for large industrial machines, and the other for the small portable-type gun welder. The tool consists of a housing that contains two special cutters shaped to the size of the standard tips being used. Turning the housing by the handles brings the cutters into play, giving a fine machined finish to the previously "mushroomed" electrode tips.

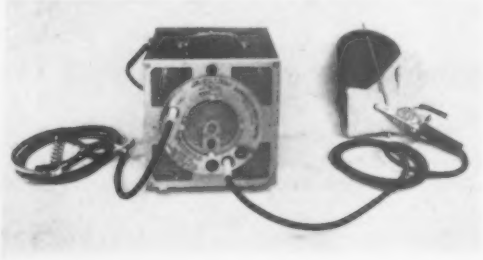
Later, it is expected that other models will be introduced capable of dressing offset and other electrode tips.

The second new product is the Fararc 200 fan-cooled toroidal arc-welding transformer (Fig. 7).

The equipment weighs 70 lb., and is contained in a steel case measuring 14 in. cube. It can readily be carried and, apart from general industrial use, is therefore, ideally suited for maintenance welding. The power factor is between 0.8 and 0.9 lagging, the figure required by electricity boards, so that no power factor correction condenser is needed, and the set can be connected direct to any 200/250 or 400/440 V. A.C. mains. There is a simple connecting box at the back of the machine for varying the input voltage.

The set has an open circuit voltage of 50 V., an added safety factor for the operator. Glass insulated wire, wrapped where necessary in glass woven cloth, impregnated with silicone varnish and baked, is used throughout for insulation. No combustible materials are used anywhere in the set and it can operate without damage at temperatures up to 180° C. the makers claim. A thermostatic switch is incorporated which cuts off the welding current but leaves the cooling fan running so that the windings rapidly cool to working temperature.

Fig. 7.—A low-voltage portable welding unit



Double Seamers for Irregular Work

A DOUBLE seamer designed for the double seaming of irregular work has been introduced by F. J. Edwards Ltd., 359-361, Euston Road, London, N.W.1. Made in five sizes with capacities from 16 in. diagonal by 20 s.w.g. to 44 in. diagonal by 14 s.w.g. mild steel, the exceptional size range makes the double seaming of water and oil storage tanks a worthwhile proposition. A further model is available for seaming the taper necks of square five-gallon oil drums. This machine has an extra long clamping stroke enabling the drum to be seamed over a horn and removed. Special features include the "Airflex" air operated clutch, brake unit and hold-down, and guards of expanded metal enclose all working parts.

Power is supplied by an electric motor, driving through vee belts direct to the drive shaft. An air cylinder is mounted above the machine, to actuate the hold-down plate, and the wormwheel is keyed to the vertical spindle which carries the pressure cams and spring loaded knock-off cam. The top arms, carrying the seaming rollers, oscillate to apply pressure on steel pins fitted to the lower arms.

The adjustment pressure is applied to the arms through steel toggle links and spring loaded slides controlled by the pressure cams.

Operation is simple. Work is placed on the chuck, and the action of closing the gate, automatically operates the hold-down and clamps the container into position. The operating treadle cannot be depressed until the gate is closed. Depressing the treadle transfers the air supply from the brake to the clutch, setting the seaming cycle in operation. The flanges of the double seam bottom and body are first curled together and then flattened. On completion of the seaming cycle, the motion is automatically stopped by the knock-off cam, and the air transferred back from the clutch to the brake. The gate can then be opened, releasing the hold-down and allowing the container to be removed from the clutch. The seaming time on the 44 in. machine is 65 seconds, with only 6 seconds for the 16 in. model. Weights range from 9 tons for the 44 in. model to 3 tons for the 16 in. model. The machines carry the familiar "Besco" trade mark.

Molybdenised Industrial Greases

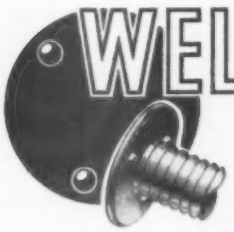
MOLYBDENISED grease for automobile and commercial vehicle chassis has been introduced by Rocol Ltd., Rocol House, Swillington, near Leeds, under the "Molyspeed" trade name.

The makers say that Molyspeed grease has 320 penetration so that it can be applied in pressure systems or in hand grease guns. It contains adequate molybdenum disulphide for general chassis work, and the particle size is compatible with the surfaces concerned.

The grease has a bentone base, is resistant to high temperatures and is water repellent; it can be used for wheel bearings and in exposed conditions.

In the United States, molybdenised chassis greases have already appeared, and much publicity was given recently to a study following a two year, two million mile vehicle road test in which wear on automobile steering assemblies was claimed to have been reduced by 38 per cent and front end suspensions by 26 per cent.

The main advantage claimed for the grease is its higher resistance to pressure, and provision of boundary lubrication between working surfaces, compared with conventional automobile greases. Scraping action does not remove the lubricant entirely and wear is inhibited or materially reduced, the makers say.



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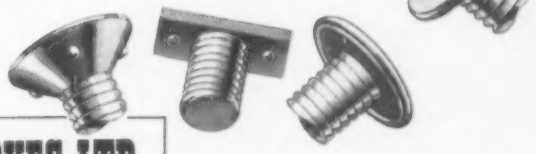

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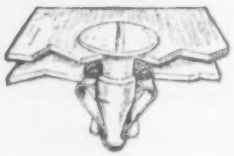

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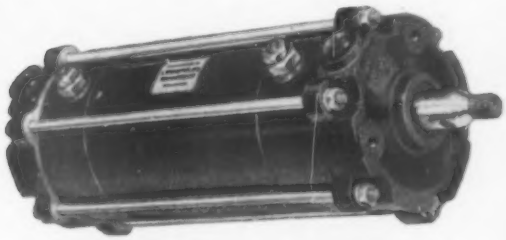

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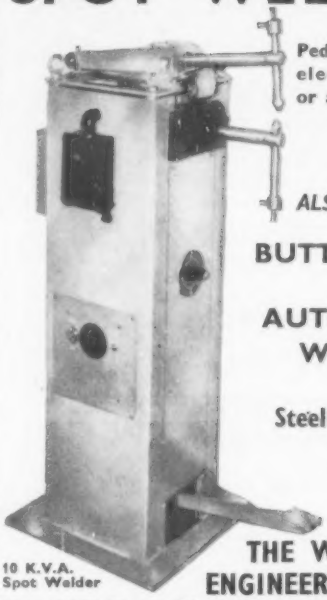

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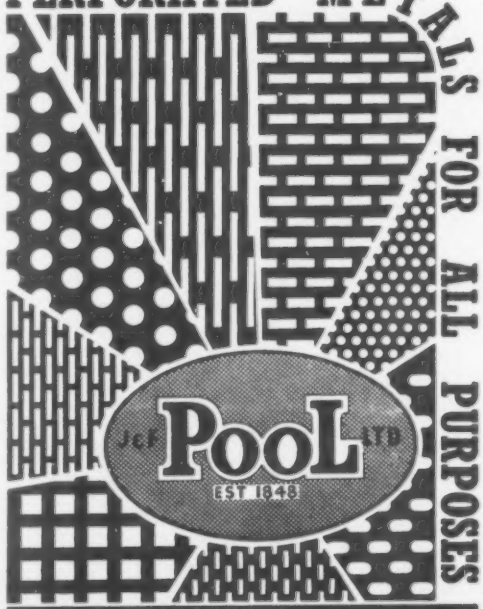

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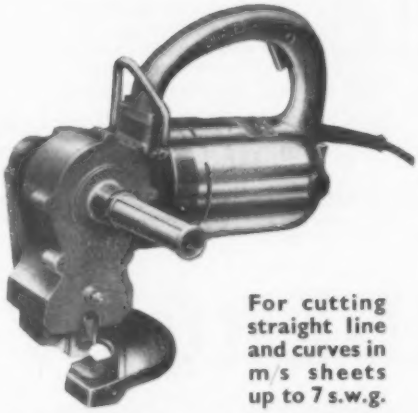

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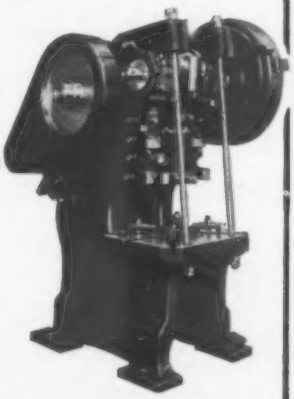

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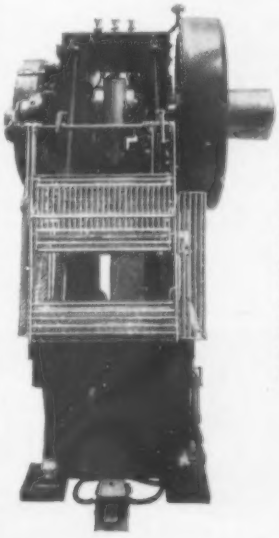

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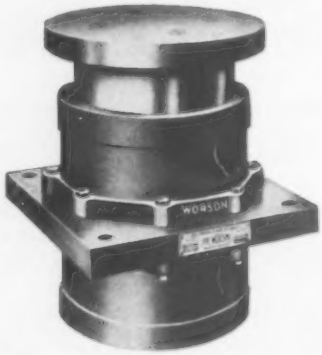

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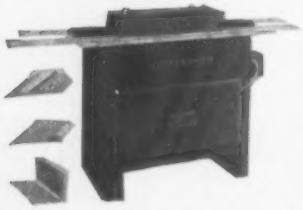

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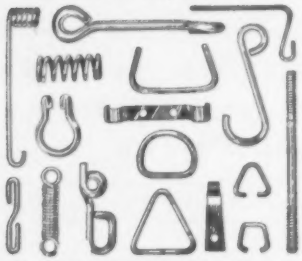

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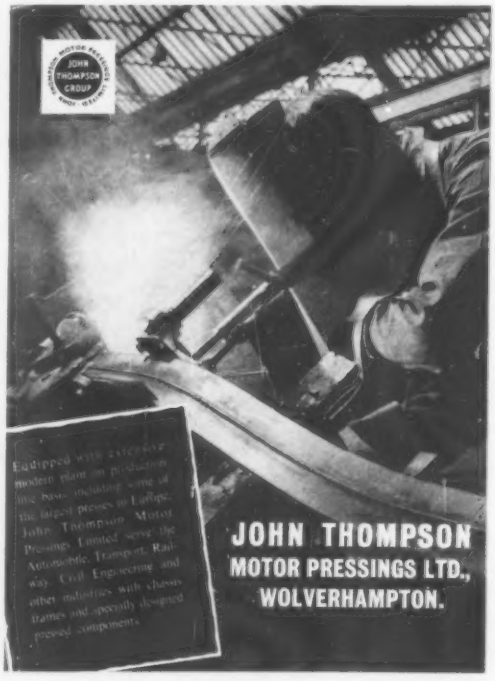
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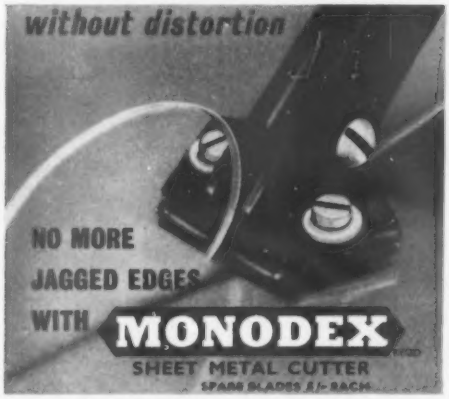

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INDEX TO ADVERTISERS

*(*denotes pages numbered in editorial sequence)*

PAGE

A.I. Electric Welding Machines Ltd. .. .. —
Aerograph-DeVilbiss Co. Ltd., The .. .. —
A.M.P. Engineers Ltd. —
A.R.O. Machinery Co. Ltd. 21
Airscrew Co. & Jicwood Ltd., The 38
Alcan (U.K.) Ltd. .. 41
Allday, William & Co. Ltd. —
Arc Manufacturing Co. Ltd. —
Ashton, N. C., Ltd. .. 80
Associated Electrical Industries Ltd. 8
Atkinson, W. B. & U., Ltd. 106
Auxiliary Rolling Machinery Ltd. 44
Aviation Developments Ltd. —
Ayrshire Dockyard Co. Ltd. 76

B.H.P. Machine Tool Co. —
Bagshawe Co. Ltd. .. —
Baldwin Instrument Co. Ltd. .. .. .. —
Ballard, F. J., & Co. Ltd. .. 42
Barns, W. & Son (Holloway) Ltd. .. .. .. —
Bartlett, G. F. E. & Son Ltd —
Batchelor, Robinson & Co. Ltd. 100
Bird, Chas. W., Ltd. .. 103
Birlec Ltd. 2*
Birmabright Ltd .. .. —
Birmetals Ltd. .. .. —
Blackman, Keith, Ltd. .. —
Bliss, E. W. (England) Ltd. 69
Bradley & Burch (Wire Components) Ltd. .. 100
Brauer, F., Ltd. 47
Brightside Foundry & Engineering Co. Ltd. .. —
British Aluminium Co. Ltd. —
British Electrical Repairs Ltd. .. .. .. —
British Federal Welder & Machine Co. Ltd. .. —
British Insulated Callender's Cables Ltd. .. .. —
British Oxygen Gases Ltd. 19, 58
British Resin Products Ltd. —
British Rolling Mills Ltd. .. —
British Timken: Division of the Timken Roller Bearing Company *Cover* iii
Broadway Equipment Ltd. —
Bromford Iron & Steel Co. Ltd., The 91
Bronx Engineering Co. Ltd. 27
Brookes (Oldbury) Ltd. .. 53
Broughton, J., & Son (Engineers) Ltd. 99
Burnett & Lewis Ltd. .. —
Burnley Aircraft Products Ltd. .. .. .. —
Butcher, Henry, & Co. .. —
Butterley Co. Ltd., The .. 73

Cam, Fred (Engineers) Ltd. 51
Cashmore, John, Ltd. .. 20
Ciba (A.R.L.) Ltd. .. —
Cincinnati Shaper Co. Ltd., The 95
Coated Metals Ltd. .. 71
Cohen, Geo., Sons & Co. Ltd. 28, 61
Comec .. .. .. —
Consolidated Pneumatic Tool Co. Ltd. .. .. —
Corfield and Buckle Ltd. .. 52
Cornercroft Ltd. 74
Corre, Alfred A., & Co. .. 105
Cowlishaw, Walker & Co. Ltd. 10
Cox & Danks Ltd. .. 104
Crosland, William, Ltd. .. —
Cruickshank, R., Ltd. .. 67

Danly Machine Specialities Inc. .. .. .. —
Davy & Utd. Eng. Co. Ltd. —

PAGE

Davy & Utd. Instruments, Ltd. .. .. .. —
Davy & United Roll Foundry Ltd. .. .. .. —
Desoutter Bros. Ltd. 36
Doncaster, Daniel, & Sons Ltd. 25
Dorman Long (Steel) Ltd. —
Dowding & Doll Ltd. .. —
Drakesons (General Metal Spinners), Ltd. 106
Ductile Sheet Stockists Ltd. 11
Duplex Electric Tools Ltd. 101

Edwards, F. J., Ltd. *Cover* iv
Effingham Steel Works Ltd. 26
Ekco Electronics Ltd. .. 63
Electrical Development Assn. .. .. .. —
Electro-Chemical Engineering Co. Ltd. .. .. —
Elgar Machine Tool Co. Ltd. 65
English Electric Co. Ltd. —
Enthoven Solders Ltd. .. 86
Expanded Metal Co. Ltd. .. 85
Expandite, Ltd. 45

Farmer Norton, Sir J., & Co. Ltd. 40
Fearnehough, W., Ltd. .. 74
Fenner, J. H., & Co. Ltd. .. 31
Firth, Thos., & John Brown Ltd. 29
Firth-Vickers Stainless Steels Ltd. .. .. —
Fleming, J. & R., Ltd. .. 80
Fletcher Miller Ltd. .. 98
Fluxite Ltd. .. .. —
Foster, Yates & Thom Ltd. 43
Fraser, Andrew, & Co. Ltd. —
Fry's Metal Foundries Ltd. 3
Fuller Electric Ltd. .. 1
Fuller, Horsey, Sons & Cassell 68

General Sheet Metal Co. (Bury) Ltd. .. .. —
Gerhardy Bros. Ltd. .. 106
Godins Ltd. 81
Grades Metals Ltd. .. 57
Granby, Paul, & Co. Ltd. .. —
Griffiths, Gilbart, Lloyd & Co. Ltd. .. .. —
Grigg, C. S. W., Ltd. 83
Guest, Keen & Nettlefolds (Midlands) Ltd. .. —
Guest, Keen & Nettlefolds (South Wales) Ltd. .. 14

Haemmerle (London) Ltd. 5
Habershon, J. J., & Sons Ltd. 2
Haddon, Thomas, & Stokes Ltd. 97
Hall Bros. (West Bromwich) Ltd. 50
Harvey, G. A., & Co. (London) Ltd. 79
Head Wrightson Machine Co. Ltd. .. .. —
Headland Works Ltd. .. 102
Hemming, J. A., Ltd. 93
Herbert, Alfred, Ltd. .. 84
Holden & Hunt Ltd. .. —
Hommel, O., Co., The .. —
Hordern, Mason & Edwards Ltd. .. .. .. —
Humphris & Sons Ltd. 15
Hutton, L. A., & Co. Ltd. .. 62

I.C.I. Ltd. (Billingham Division) 82
I.C.I. Ltd. (Metals Division) —
Imperial Aluminium Co. Ltd. 6, 7
Incandescent Heat Co. Ltd —
Individual Tools Ltd. .. —
Ingersoll Locks Ltd. .. 101
Instrument Screw Co. Ltd. —

PAGE

Johnson, Rd., Clapham & Morris Ltd. .. .. —
Johnson, Matthey & Co. Ltd. —
Jones & Attwood Ltd. .. 63
Jones, E.H. (Machines Tools) Ltd. .. —

Keeton, Sons & Co. Ltd. .. 54
Kendrick, George, Ltd. .. 66
Kestner Evaporator & Engineering Co. Ltd. .. —
Kieserling, Th., & Albrecht —
Kimbell Machine Tools Ltd. —
Kimber, H., Ltd. 107
King, William, Ltd. .. 64
Kingsland Eng. Co. Ltd. .. —
Kubach, F. W., Ltd. —

Lancashire & Corby Steel Manufacturing Co. Ltd. .. 34
Lee, A., & Sons Ltd. 72
Lewispoint Holdings Ltd. —
Lindemann Kommanditgesellschaft .. .. —
Lockers (Metal Perforators) Ltd. 48
Loewy Engineering Co. Ltd. .. .. .. —
London Aluminium Co. Ltd. The .. .. .. —
London Tinners Ltd. 102
Lyseght, John, (Services) Ltd. 55

Macready's Metal Co. Ltd. 42
Marshall Richards Machine Co. Ltd. 96
Mercer, Samuel, & Co. .. 37
Metal Sections Ltd. .. —
Midland Saw & Tool Co. Ltd. —
Milne, C. S., & Co. Ltd. .. 13
Moon Bros. Ltd. .. .. —
Morleys (Birmingham) Ltd. 46
Mountford Frederick (B'ham) Ltd. .. .. —
Murex Welding Processes Ltd. 83*

Neill, James, & Co. (Sheffield) Ltd. 48
Nettlefold & Moser Ltd. —
Nichols, J., & Son Ltd. .. 104
Nicholls, Samuel .. .. —

Oddie Bradbury Cull Ltd. 98
Oliver Machinery Co. Ltd. 12, 103, 104
Orcone Co. Ltd. 76

Padley & Venables Ltd. .. 4
Parker (Toggles) Ltd. 102
Patentools Ltd. 107
Pearson Panke Ltd. 33, 49
Pearson Machine Tool Co., Ltd. 104
Pels, Henry, & Co. Ltd. .. 92
Pinchin, Johnson & Associates Ltd. 17
Plant Inspection & Control Ltd. .. .. .. —
Pistol, F. J. Ltd. .. .. —
Polarcold Ltd. .. .. —
Pool, J. & F., Ltd. 100
Press & Shear Co. Ltd. .. 39
Press Equipment Ltd. .. 77
Press Guards Ltd. .. —
Price Machine Guards Ltd. 101

Rapp, Leo, (Steel) Ltd. 104
Ratcliff J. F. (Metals) Ltd. 32
Revolvo Ltd. 67
Rhodes, J., & Sons Ltd. —
Robertson, W. H. A., & Co. Ltd. 70
Roberts, Sparrow & Co. Ltd. 22
Rockwell Machine Tool Co. Ltd. *Cover* ii, 87, 89

PAGE

Rollet, H., & Co. Ltd. .. —
Rowley, F.R. (Technical Supplies) Ltd. .. .. —
Rowntree Graham & Co. Ltd. 106
Rustless Iron Co. Ltd. .. —

S. & D. Rivet Co. 62
Sahlin Engineering Co. Ltd. —
Sanderson Bros. & Newbould Ltd. .. .. .. —
Saturn Industrial Gases Ltd. —
Sciaky Electric Welding Machines Ltd. 35
Scottish Machine Tool Corporation Ltd. 56
Selecta Power Tools Ltd. —
Shaw, J., & Sons (Salford) Ltd. 59
Shawe Metal Spinning Works 105
Shell Chemical Co. Ltd —
Shepperd Metal Productions Ltd. 106
Shimwell & Co. Ltd. .. 75
Siemens-Schuckert (Great Britain) Ltd. 90
Simmonds Aerocessories Ltd. .. .. .. —
Simpson, J., & Son (Engineers) Ltd. .. .. —
Skerne Works Ltd., The .. 107
Smith & McLean Ltd. .. 52
Speed Tools Ltd. 65
Spencer, H. F., & Co. Ltd. 96
St. Ann's Sheet Metal Co. —
Stainless Recoveries Ltd. .. 103
Steel Company of Wales Ltd., The 18
Steel Stampings Ltd. .. 99
Steel, Peech & Tozer Ltd. —
Steels & Busks Ltd. .. —
Stordy Engineering Ltd. .. —
Suffolk Iron Foundry (1920) Ltd. 98
Summers, John & Sons Ltd. 16
Sweeney & Blocksidge (P.P.) Ltd. 101
Symmons, Edward, & Partners .. .. —

Tangyes Ltd. 30
Taylor & Challen Ltd. .. 81*
Taylor Rustless Fittings Ltd. 105
Taylor Stainless Metals Ltd. 68
Tedson, Thornley & Co. Ltd. .. .. .. —
Thomas, Richard & Baldwins Ltd. .. *Cover* i, 9, 94
Thompson, John (Dudley) Ltd. .. .. .. —
Thompson, John, Motor Pressings Ltd. 106
Time Recorder Supply & Maintenance Co. Ltd. .. 103
Tucker, Geo., Eyelet Co. Ltd. 32

Udal, J. P., Ltd. 50
United Steel Companies, Ltd —

Vaughan Associates Ltd. 78
Veeder-Root Ltd. 75

Walker, James, & Co. Ltd. —
Ward, Thos. W., Ltd. 103
Watson, Saville & Co. Ltd. 64
Wellman Smith Owen Engineering Corpn Ltd. 88
Welsh Metal Industries Ltd. —
Westminster Engineering Co. Ltd. 100
Whitehead Iron & Steel Co. Ltd. 60
Wickman Ltd. .. .. —
Wiggin, H., & Co. Ltd. .. —
Wilbraham & Smith Ltd. .. 24
Wilkes, A. H., & Co. .. —
Wilkins & Mitchell Ltd. 23
Willmott Taylor Ltd. .. —
Wolf Electric Tools Ltd. .. —
Worson Die Cushions Ltd. 102
Wright, Bindley & Gell Ltd. 97

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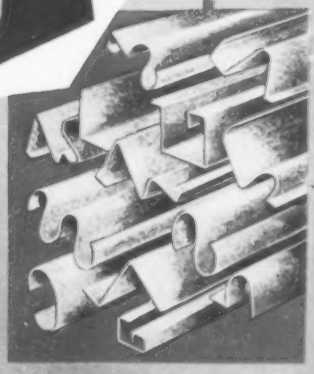

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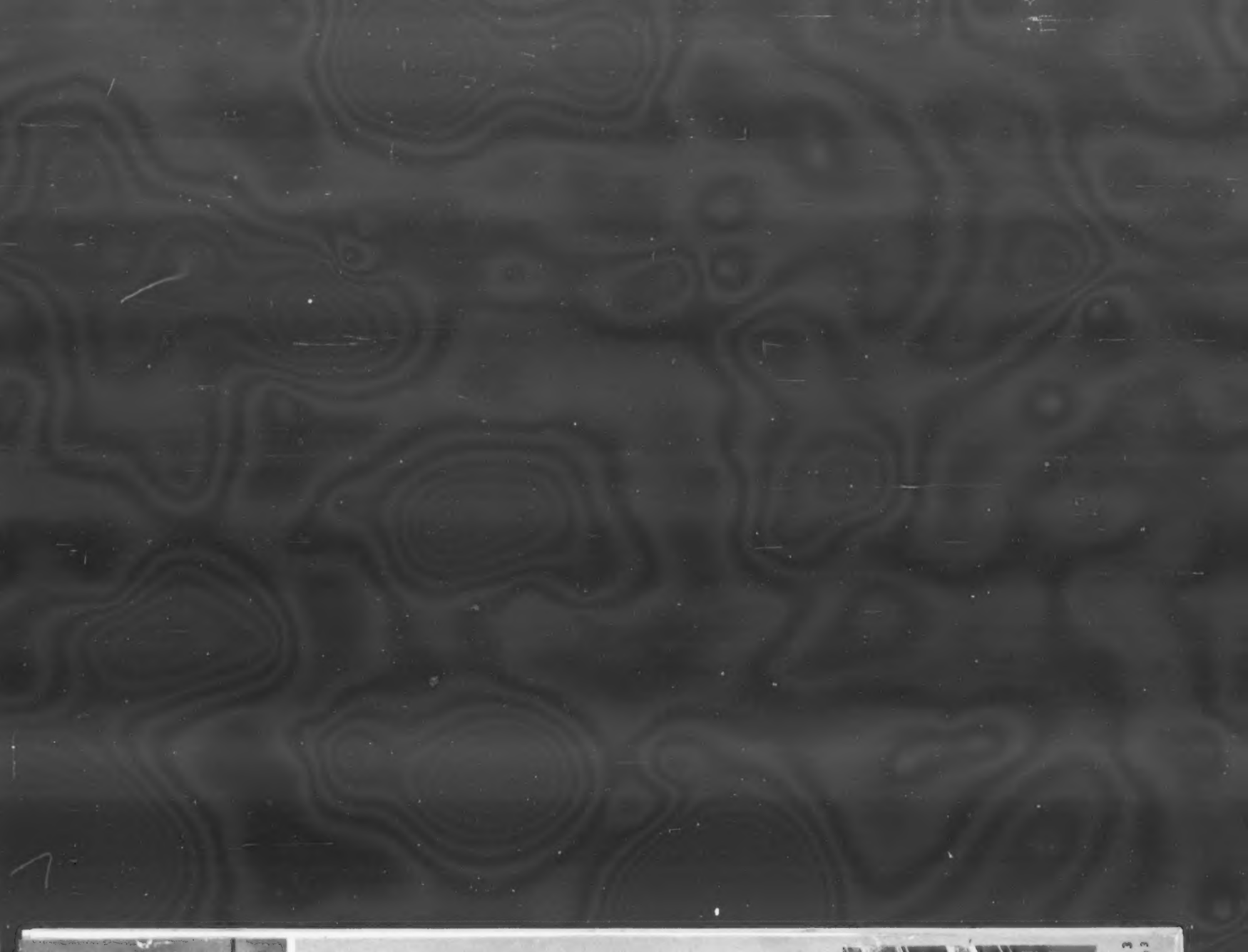
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